

technology." Based on this survey, which was planned to include all the geographical areas known to have relatively stringent regulations, it was concluded that no installed engines were currently subjected to any emission limitations other than for visible emissions. The only exception was the 140 lb/hr rule for NO_x from new engines in Southern California (see Section 2.2). Since most uncontrolled engines under 4000 hp can meet this regulation, and since no larger engines are known to have been installed in that area recently, that rule did not lead us to any examples of field-installed controlled engines. Therefore, the evaluation of control technology was to be based upon published emission data and data obtained from engine manufacturers.

Since no installed engines were found which could be cited as examples of units with "best" controls, no emission tests were conducted. However, a San Jose, California sewage plant (San Jose Water Pollution Control Plant) was visited to give a better understanding of the operational flexibilities available to users of large-bore engines.⁽⁴⁾ This plant had six gas engines and six dual-fuel units.

The direct contacts were supplemented by visits to the manufacturers listed in Table A-2. The purpose of these visits was to obtain more information concerning the status of their R&D efforts in emission reductions, their experience with the commonly proposed control technologies, their estimates of the cost and time required to incorporate such controls on their engines, and the importance of the stationary market place in their total sales.

The group meetings that were held with representatives of engine manufacturer's associations are listed in Table A-3. In general, the meetings served to inform these representatives of the purpose of performance standards for new sources, the process of developing a standard, including

Table A-2. ENGINE MANUFACTURERS VISITED DURING
PREPARATION OF THE STANDARDS SUPPORT DOCUMENT

Company	Location	Date Visited	Category	Reason for Selection
Caterpillar, ^a Tractor Co.	Peoria, IL	8/27/74	Medium Bore	Make precombustion chamber engines; have published several technical papers on emissions
Cummins, ^a Engine Co.	Columbus, IN	8/28/74	Medium Bore	One of best known manufacturers of truck engines; have published several technical papers on emissions
Enterprise Engine Div.	Oakland, CA	7/18/74	Large Bore	One of largest manufacturers in large-bore category; close to Aerotherm/Acurex
Waukesha Motor Co.	Waukesha, WI	7/25/74	Medium and Large Bore	Market widest range of engine sizes
Worthington	Buffalo, NY	9/1/74	Large Bore	Manufacturer of large-bore gas engines

^a Aerotherm staff engineer accompanied by EPA Project Officer

TABLE A-3. MEETINGS WITH ENGINE MANUFACTURERS ASSOCIATIONS

Date	Association	Meeting Location	EPA/Aerotherm Attendance	Report Date
8/28/74	Diesel Engine Manufacturers Assoc. (DEMA)	EPA (Durham, NC)	EPA	9/3/74
11/21/74	Engine Manufacturers Assoc. (EMA)	EPA (Durham, NC)	EPA and Aerotherm ^a	11/26/74
4/17-18/75	Engine Manufacturers Assoc. (EMA)	Aerotherm/Acurex (Mountain View, CA)	Aerotherm	4/22/75 (2 reports)
6/10/76	American Gas Assoc. ^b	EPA (Durham, NC)	EPA	6/26/75
5/28/76	American Gas Assoc.	EPA (Durham, NC)	EPA	4/30/76
5/4/76	Colt Industries	EPA (Durham, NC)	EPA	5/18/76
5/12/76	Engine Manufacturers Assoc. (EMA)	EPA (Durham, NC)	EPA and Aerotherm	5/18/76
5/13/76	Diesel Engine Manufacturers Assoc. (DEMA)	EPA (Durham, NC)	EPA and Aerotherm	5/18/76
5/14/76	General Motors Corporation	EPA (Durham, NC)	EPA and Aerotherm	5/18/76

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^aMeeting report prepared by Aerotherm

^bSupervisory committee for Project PR-15-61, "NO_x Research". Accompanied by staff from Southwest Research Institute

informal and formal opportunities for input from all interested parties, and the reasons for considering that stationary engines should be regulated by performance standards. The meetings were also used to request information of a general nature, such as data on the effects of variable ambient conditions (temperature, humidity, etc.) on emissions, appropriate test cycles (based on their knowledge of engine usage patterns) possible engine operating parameters that are known to be directly and uniquely related to emission levels and could be used in lieu of emission testing for compliance monitoring. As is also shown in Table A-3, representatives of three industry organizations met with EPA and commented on the draft SSEIS after it was presented at the March 1976 NAPCTAC meeting in Washington, D.C.

Several meetings were held between Acurex/Aerotherm and EPA staff during the development of the draft document. In the first half of 1978, two public meetings were held to discuss the proposed standards, one before the NAPCTAC committee and the other with the engine manufacturers. These are listed in Table A-4 along with a brief description of the purpose, conclusion, and any redirections.

In early 1978, Acurex was assigned the responsibility for writing the rationale chapter of the SSEIS, the preamble to the standard, and the regulation itself. In addition, Acurex was also assigned responsibility for establishing monitoring and emission testing procedures. This work was performed with the advice and review of EPA/OAQPS.

Review of the original draft Standards Support Document and Environmental Impact Statement by the Industrial Studies Branch, Emission Standards and Engineering Division, OAQPS was completed on December 20, 1975, and the final draft was forwarded to the Industrial Studies Branch by Acurex/Aerotherm on March 17, 1976. The revised draft SSEIS was reviewed by the

Standards Development Branch, Emission Standards and Engineering Division,
OAQPS by March 1978 and a final SSEIS was forwarded to the Standards
Development Branch by Acurex/Aerotherm in July 1978.

TABLE A-4. MAJOR PROJECT REVIEWS AND DECISION-RELATED COMMUNICATIONS

Date	Format ^a	Purpose	Conclusions/Redirections
7/23-24/74	EPA-Aerotherm meeting; report prepared by EPA (August 5, 1974)	Review proposed Work Plan prepared by Aerotherm	Guidance provided to Aerotherm on procedural matters, including allowable and effective methods of collecting emission and marketing data, precautions to consider when requesting and evaluating user or manufacturer supplied emission data, air pollution control agencies that might regulate engines, protocol of interacting with users and manufacturers, desire to determine if one standard could be set rather than separate standards for each engine type and fuel, etc. Decision made to collect as much information as possible by telephone and individual follow-up letter rather than by questionnaire canvas of whole industry.
9/19/74	Memorandum (Industrial Studies Branch to Office of Enforcement and General Counsel)	Request legal opinion on time-phased standards, ability to treat standards on total hydrocarbons as standards on a criterion pollutant	See December 18, 1974 for response
9/27/74 10/1/74	Telephone conversations (EPA Project Officer and Aerotherm) ^b	Relay decision by EPA (Industrial Studies Branch) to solve problem of insufficient data by sending "Section 114" letters to selected engine manufacturers	Aerotherm to provide EPA with list of manufacturers who had not yet been contacted formally and were believed to have useful information (sent October 10, 1974 - October 24, 1974)

^aAll meetings held in EPA offices, Durham, NC unless otherwise noted.

^bMany other telephone conversations occurred during which procedural details were discussed. These are not documented here.

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TABLE A-4. Continued

Date	Format ^a	Purpose	Conclusions/Redirections
12/18/74	Memorandum (Office of General Counsel to Industrial Studies Branch)	Response to ISB memos of September 19 and November 27, 1974	Mandatory certification and recall program not permitted under Section 111, Section 114 binding on manufacturers for emission and cost to control data, effective date of standard need not be proposal date, and a standard on total hydrocarbons would probably not be a standard on a noncriterion pollutant
2/10-11/75	EPA-Aerothrm meeting; reports prepared by Aerothrm (Feb. 13, 1975) and EPA (Feb. 25, 1975)	To present status of project to ESED staff and discuss alternatives considered in major decision areas (selection of affected facilities, selection of pollutants for control, and proposed format for the standard)	Retabulate uncontrolled and controlled data by engine and fuel type. Consider need to propose separate standards by engine type and fuel based on economic impact. Examine technological and cost impact of subjecting standby engines to standards. Include some additional factors in rationale for proposed pollutants to be controlled and format of standard
3/28/75	Meeting with Director, Emission Standards and Engineering Division, OAQPS (ISB report dated April 3, 1975)	Report on status of project to Division, present recommendations to limit source category to engines over 500 hp, and identify anticipated problems	Continue to develop standards of performance for stationary engines over 500 hp. A possible result of the standard development study may be to recommend R&D program to Industrial Environmental Research Laboratory, RTP.
4/4/75	Letter (EPA Project Officer to Aerothrm)	Report on March 28 meeting with Director, ESED	Reorganize format of draft document to minimize discussion of engines less than 500 hp and limit the development of standards to engines over 500 hp.
4/30/75	Letter (EPA Project Officer to Aerothrm)	Transmit additional instructions on format of report	Incorporate minor changes as requested
7/30/75	Letter (EPA Project Officer to Aerothrm)	Indicate changes in drafts of reports submitted to Project Officer for review	Limit standard to engines greater than 350 CID/cyl

^aAll meetings held in EPA offices, Durham, NC unless otherwise noted.
^bMany other telephone conversations occurred during which procedural details were discussed. These are not documented here.

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TABLE A-4. Continued

Date	Format ^a	Purpose	Conclusions/Redirections
3/17/76	Meeting of National Air Pollution Control Techniques Advisory Committee (NAPCTAC) in Washington, D.C.	To present to NAPCTAC and industry the recommended standards of performance for stationary reciprocating IC engines and the background information leading to this recommendation	Industry to review the Standards Support and Environmental Impact Statement and meet with EPA in May 1976
5/13/76	Meeting with the Diesel Engine Manufacturers Association (DEMA)	DEMA presented their comments and recommendations on emission standards for stationary IC engines	Since substantially more emission data were now available, "Section 114" letters were to be prepared to request this information from large-bore engine manufacturers
6/16/76	"Section 114 Request for Information" letter sent to large-bore (>350 CID/cyl) engine manufacturers	To obtain additional emissions data, market information, and control cost information	Acurex/Aerotherm to incorporate additional data in the SSEIS. Aerotherm received responses from "114" letters on 8/20/76
9/17/76	EPA-Aerotherm meeting; report prepared by EPA (September 29, 1976)	To discuss applicability of new data (received from "114" responses) to the data base of the draft SSEIS, and to discuss revisions of the draft SSEIS	Aerotherm plan for incorporating additional data approved
11/17/76	EPA-Aerotherm meeting at Mountain View, CA	To discuss expansion and refinement of existing emissions data base	Aerotherm to document work concerning additional data, ambient corrections measurement procedures, and field applicability of control techniques in a supplemental report
1/28/77	Engine Manufacturers Association (EMA)-Aerotherm meeting at Mountain View, CA	To update EMA members on Aerotherm's revisions to draft SSEIS	EMA Stationary Engine Committee to advise Aerotherm of suggestions for a definition of engines to be affected by NSPS, engine manufacturer's comments on EPA's mobile source test procedures, and influence of ambient conditions on emission measurements. (Response received 3/21/77.)

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^aAll meetings held in EPA offices, Durham, NC unless otherwise noted.
^bMany other telephone conversations occurred during which procedural details were discussed. These are not documented here.

TABLE A-4. Continued

Date	Format ^a	Purpose	Conclusions/Redirections
2/24/77	EPA-Aerothrm meeting; report prepared by Aerothrm (3/1/77)	Status report on revisions to SSEIS	EPA to meet with Aerothrm to discuss alternatives for determining an emission standard
3/9-10/77	EPA-Aerothrm meeting at Mountain View, CA; report prepared by Aerothrm (3/15/77)	To review emission data compilation and to discuss the development of alternative control strategies	Controlled levels to be determined by applying a percentage reduction demonstrated by viable control technologies to sales-weighted uncontrolled averages
3/24/77	Letter (EPA Project Officer to Aerothrm)	Advise Aerothrm of format revised draft SSEIS	Aerothrm provided EPA with tables of contents for chapters of SSEIS undergoing major revision (letter of 4/15/77)
9/21-22/77	EPA-Aerothrm meeting; report prepared by Aerothrm (9/28/77)	To review revisions being made to draft SSEIS; in particular, the alternative control techniques and the definition of affected facilities	Aerothrm to work with EPA contractor in revising economic impact analysis. Aerothrm to deliver revised draft SSEIS by November 1977, economic impact analysis to be completed in December 1977.
11/15,16/77	EPA-Aerothrm meeting at Mountain View CA; report prepared by Aerothrm (11/21/77)	To discuss change in Aerothrm's scope of work and review draft SSEIS	Aerothrm to write rationale chapter, preamble and regulation with assistance from EPA; Chapters 3, 4, 5, 6, 7, and Appendices A and C approved.

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^aAll meetings held in EPA offices, Durham, NC unless otherwise noted.
^bMany other telephone conversations occurred during which procedural details were discussed. These are not documented here.

TABLE A-4. Concluded

Date	Format ^a	Purpose	Conclusions/Redirections
1/25/78	EPA-Aerothrm meeting to discuss economics impact chapter	General discussion of development of economics impact chapters and discussion of future schedule	Future schedule to prepare for NAPCTAC meeting defined
4/5-6/78	EPA-Aerothrm meeting at OAQPS and Acurex presentation to NAPCTAC	Prepared for NAPCTAC presentation to be given the following day in Washington	Engine manufacturers presented viewpoints and requested future meeting to discuss problem areas
4/26-29/78	EPA-Aerothrm manufacturers meeting to discuss questions arising from NAPCTAC	Receive from manufacturers their comments and viewpoints on standards recommended at NAPCTAC	Letter drafted and sent to manufacturers specifically requesting additional information to help reevaluate suggested standards
5/30/78	EPA-Aerothrm meeting to discuss rationale chapter	To discuss organization of rationale chapter and modification of proposed standards based on data received from manufacturers	Standards to be revised based on operational data supplied by manufacturers and statistical analysis of impact of standard
6/29-30/78	EPA-Aerothrm meeting on rationale, preamble regulation	Discuss final revisions to rationale chapter, preamble and regulation-revision to test method and monitoring discussed	Set schedule for delivery of draft of SSEIS to Steering Committee test method to be based on Method 20

^aAll meetings held in EPA offices, Durham, NC unless otherwise noted.^bMany other telephone conversations occurred during which procedural details were discussed. These are not documented here.

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REFERENCES FOR APPENDIX A

- (1) Diesel and Gas Turbine Worldwide Catalog. Diesel and Gas Turbine Progress. Milwaukee, Wisconsin. 39. 1974.
- (2) McGowin, C. G., Stationary Internal Combustion Engines in the United States. EPA-R2-73-210. April 1973.
- (3) Roessler, W. V., A. Muraszew, and R. D. Kopa. Assessment of the Applicability of Automotive Emission Control Technology to Stationary Engines. EPA-650/2-74-051. July 1974.
- (4) Offen, G. R. (Acurex/Aerotherm). Trip Report. San Jose Water Pollution Control Plant. (Interoffice Memorandum to F. E. Moreno, Acurex/Aerotherm) June 5, 1975.

APPENDIX B

Agency Guidelines for Preparing Regulatory Action Environmental Impact Statements (39 FR 37419)

Location Within the Standards Support and Environmental Impact Statement

1. Background and Description of Proposed Action

Summary of Proposed Standards

The standards are summarized in Chapter 1.

Statutory Basis for the Standard

The statutory basis for the standard is given in Chapter 2.

Facility Affected

A description of the facility to be affected is given in Chapter 3.

Process Affected

A description of the process to be affected is given in Chapter 3.

Availability of Control

Information on the availability of control technology is given in Chapter 4.

Existing Regulations at State
or Local Level

A discussion of existing regulations on the industry to be affected by the standard is included in Chapter 3, Section 2.3.

2. Alternatives to the Proposed Action

Environmental Impacts

Environmental effects of delaying the standards are discussed in Chapter 7, Section 6.

Costs

The costs of alternative control techniques are discussed in Chapter 8, Sections 2, 3, and 4.

Agency Guidelines for Preparing
Regulatory Action Environmental
Impact Statements (39 FR 37419)

Location Within The Standards
Support and Environmental
Impact Statement

3. Environmental Impact of
Proposed Action

Air Pollution

The air pollution impact of the standards is considered in Chapter 7, Section 1.

Water Pollution

The water pollution impact of the standards is discussed in Chapter 7, Section 2.

Solid Waste Disposal

The solid waste disposal impact of the standards is discussed in Chapter 7, Section 3.

Energy

The energy impact of the standards is considered in Chapter 7, Section 4.

Other

The environmental impacts related to noise and thermal pollution are discussed in Chapter 7, Section 5.

APPENDIX C

EMISSION SOURCE TEST DATA

C.1 DATA

This appendix tabulates all the quantitative emission data that were obtained for large-bore engines during the preparation of the Standard Support and Environmental Impact Statement. These data include emission and fuel consumption rates for both baseline and controlled conditions. Since no controlled stationary reciprocating engines are known to exist in field installations, all the data were obtained from engine manufacturers, who reported on the results of experiments in their laboratories. In some cases, these data have been made public, but in most instances they were received directly from the manufacturer as a private communication.

Large stationary engines are expensive to purchase and operate, and controlling their emissions has not previously received high priority. For these reasons, virtually no government, commercial (other than manufacturers), or university test labs have obtained one or reported on possible emission reduction technologies that might apply to currently produced engines.^{1/} However, Southwest Research Institute has a two-cylinder version of an Electromotive Division (EMD) locomotive engine which they have used for emission control

^{1/}Recently attained improvement in thermal efficiency by operation at increased pressure and temperature precludes the use of emissions data from older engines.

investigations(1). This engine is no longer being produced by EMD, although it represents most of the current locomotive engine population. According to data supplied by EMD, this model's successor has higher baseline NO_x emissions (14.7 g/hp-hr for the newer model compared to 9.1 g/hp-hr for the older two-cylinder version at rated conditions) and lower total hydrocarbon emissions (0.49 g/hp-hr compared to 0.92 g/hp-hr). These differences are consistent with the increased efficiency of the engine. New EMD data (Engines 17, 18 and 19) showed the same trends with the application of controls that the SwRI engine data did; consequently, SwRI's results are not included here.

The data include engines which are currently marketed for stationary application and also several units used in mobile and marine applications. However, data from installed engines were excluded because these older units are not the same as those new units that would have to meet any promulgated standards.

NO_x emissions were measured using one of the four procedures and several kinds of NO_x analyzers. These procedures and instruments are identified and discussed in Section 4.2 for each large-bore engine manufacturer that reported emissions data. NO_x emissions were measured by either a nondispersive analyzer or a chemiluminescent instrument with a thermal reactor to convert the NO₂ to NO. Results were reported as grams NO₂.

Nondispersive infrared analyzers (NDIR) were used to measure the CO concentrations. Since the exhaust was "dry," interference due to water was negligible. When CO₂ and oxygen were measured, an NDIR analyzer was used for the CO₂ and a paramagnetic analyzer for the oxygen. These constituents were generally sampled to check the other measurements or to calculate total exhaust flowrate if the inlet air flowrate into the engine was not measured.

Total hydrocarbon emissions were measured in a heated flame ionization detector (FID). The tubing and FID were maintained at 375°F to prevent condensation, and hence, removal of any of the heavier hydrocarbons. Therefore, these measurements were made on a "wet" basis because the water had not been removed. The results were corrected for the water content to make them correspond with the NO_x and CO data. Since the exhaust from gas engines does not contain these easily condensable heavy hydrocarbons, the lines and FID were not heated in some tests on gas engines. In addition to these total hydrocarbon measurements, a few manufacturers measured the normethane emissions from gas and dual-fuel engines. Their methods and results are discussed in Appendix C.4.

Four engine manufacturers (Colt, Cooper, DeLaval, and White Superior), who are members of the Diesel Engine Manufacturers Association (DEMA), measured emissions data according to the measurement procedure published by DEMA(2). Although the DEMA methodology suggests that a three-mode cycle be used for engines that will be used at constant speed (essentially all stationary applications) and a six-mode cycle for units that will see variable speed duty (marine and locomotive), most of the results reported here were collected prior to the final publication of the DEMA procedures. Hence, emissions were usually measured only at rated load and speed. However, some engines were also tested at selected part load and speed points. These additional data points appear on the following tables (see Tables C-1 through C-82) as derated operating points or as variations in the engine operating speed.

Several of the engines (Engine Nos. 30 through 33) at the low horsepower end of the large engine category were tested according to the

California 13-mode cycle. However, only the emissions at rated conditions were used here since the 13-mode automotive cycle does not represent stationary engine usage, and no relationship was found between emission rates based on the average of the California 13-mode cycle and those at rated conditions.

The data were submitted in either tabular or graphical form. In the latter case the data points were clearly marked and could be converted readily to the tabular format for inclusion in this appendix. Other operational data were either supplied by the manufacturer or obtained from product information literature. Fuel consumption values in all cases are specific to the test that measured the emissions and are not from general product specifications.

Fuel consumption was generally reported in terms of lb/hp-hr for diesels and Btu/hp-hr for gas and dual-fuel engines. Conversion to kcal/hp-hr was accomplished using a lower heating value of 18,320 Btu/lb for diesel fuel and a units conversion of 0.252 kcal/Btu. All data presented in this document on emissions from diesel and dual-fuel engines are based on operation with No. 2 distillate (diesel oil).

In several cases, a number of tests were performed on the same engine, and consequently an uncontrolled value was reported for each control technique. In all cases the uncontrolled level precedes any controlled data reported for a particular test. In the tabulations which follows, a separate table is used for each engine, and the data are printed in numerical order based on engine code number. These data come from References 1 through 22. These references are noted on the tables except where an engine manufacturer has requested anonymity.

TABLE C-1. EMISSIONS SOURCE TEST DATA -- ENGINE 1

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				HC ^a	CO	HC ^a							
1	Gas	82.5 bmeep (psi)	Uncontrolled	15.1	0.3	1.9	--	7127	--	62.5	--	--	--
		1080 hp	Derate power 12%	3.84	0.72	2.68	--	7311	2.6	--	--	--	--
		300 rpm	Air-to-fuel ratio inc. 25%	14.66	0.22	2.14	--	7223	1.3	68.2	--	--	--
		2 cycle, injected	Inc. to 330 rpm at constant power	6.41	0.41	2.25	--	7192	0.9	68.0	--	--	--
		Blower scavenged BS (130°F)	Inc. to 400 rpm at constant power	1.25	1.45	0.87	--	8291	16.3	--	--	--	--
		8 cylinder	Dec. intake manifold air temperature 30°F	9.6	0.34	2.04	--	7112	-0.2	61.9	--	--	--
		14" bore x 14" stroke	Inc. exhaust back-pressure	9.53	0.3	2.16	--	7673	7.7	--	--	--	--
		2115 in ³ /cyl	3° ignition retard	14.25	0.35	2.0	--	7378	3.5	--	--	--	--
		135 hp/cyl	6° ignition retard	12.75	0.35	2.26	--	7496	5.2	59.0	--	--	--
		Cooper-Bessemer Company	9° ignition retard	13.5	0.35	2.0	--	7751	8.7	--	--	--	--
		Model GMVA-8	6° ignition retard and decrease intake manifold air temp. 30°F	10.63	0.32	2.08	--	7572	6.2	58.8	--	--	--
		Ref. 3, 4											
			Above plus air-to-fuel ratio 13% inc.	8.73	0.31	2.19	--	7654	7.4	66.2	--	--	--

^aTotal hydrocarbons

^bBrake specific fuel consumption

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TABLE C-2. EMISSIONS SOURCE TEST DATA -- ENGINE 2

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
2	Gas	111.4 bmep 1600 hp 330 rpm 2 cycle injected Turbocharged Aftercooled (130°F) 8 cylinder 14" bore x 14" stroke 2155 in ³ /cyl 200 hp/cyl Cooper-Bessemer Company Model GMVH-8 Ref. 3	Uncontrolled Derate power 25% 4° ignition retard Inc. air-to-fuel ratio to 10% excess air Dec. intake manifold air temperature 30°F 4° ignition retard & deceleration intake manifold air temp. by 30°F Above plus inc. air-to-fuel ratio to 10% excess air	13.42	0.2	1.45	--	6775	--	54.7	--	--	--
				4.59	0.3	1.76	--	7186	6.1	51.2	--	--	--
				11.76	0.27	1.47	--	7069	4.3	52.1	--	--	--
				7.43	0.22	1.69	--	6873	1.4	59.0	--	--	--
				8.27	0.21	1.58	--	6810	0.5	54.8	--	--	--
				8.09	0.22	1.57	--	7137	5.3	52.2	--	--	--
				4.59	0.22	1.63	--	7210	6.4	56.5	--	--	--

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-3. EMISSIONS SOURCE TEST DATA -- ENGINE 3

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
3	Diesel	141 hp bmeep 2475 hp 900 rpm 2 cycle Turbocharged Aftercooled 12 cylinders 9.06" bore x 10" stroke 645 in ³ /cyl 206 hp/cyl ElectroMotive Division, General Motors Corporation Model 21-645E-3 Ref. 5	Uncontrolled ^c	12.61	1.19	0.38	--	--	--	--	--	--	--

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cRated load

TABLE C-4. EMISSIONS SOURCE TEST DATA -- ENGINE 4

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _f ^a							
4	Gas	175 bmep	Uncontrolled	12.75	--	4.26	--	6374	--	32.5	--	--	--
		3240 hp	Derate power 14%	12.14	--	3.38	--	6426	0.8	31.1	--	--	--
		514 rpm	4° ignition retard	12.43	--	2.63	--	6464	1.4	32.0	--	--	--
		4 cycle	Inc. air-to-fuel ratio to 10% excess	8.57	--	7.3	--	6520	2.3	34.7	--	--	--
		Turbocharged	air										
		Aftercooled (to 100°F)	Dec. intake manifold air temp. by 20°F	11.62	--	5.27	--	6372	-0.03	32.7	--	--	--
		12 cylinder	4° ignition retard	8.18	--	4.76	--	6593	3.4	33.8	--	--	--
		13.5" bore x 16.5" stroke	& inc. air-to-fuel ratio to 10% excess										
		2361 in ³ /cyl	air										
		270 hp/cyl	Above plus dec. in intake manifold air temp. by 20°F	7.5	--	4.76	--	6600	3.5	33.8	--	--	--

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cSmoke measured with a Bacharach Model RXB smoke meter (filter-type instrument)

TABLE C-5. EMISSIONS SOURCE TEST DATA -- ENGINE 5

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				HC _T ^a	CO	HC _T ^a							
5	Diesel	200 bhp	Uncontrolled	10.99	3.85	0.13	4	6677	--	32.6	85	100	--
		4300 hp	Derate power 38%	10.30	1.33	0.32	2	7000	4.8	--	--	--	--
		600 rpm	4° ignition retard	9.31	4.45	0.21	4	6732	0.8	33.1	100	100	--
		4 cycle	Air-to-fuel ratio	11.05	3.06	0.18	4	6596	-1.2	33.7	--	--	--
		Turbocharged	with 2% excess air	10.54	3.82	0.17	4	6583	-1.4	33.1	--	70	--
		Aftercooled (to 130°F)	Dec. intake manifold air temp. by 30°F	10.2	4.25	0.16	4	6643	-0.5	32.5	--	--	--
		12 cylinder	Induct water at 0.17 water-to-fuel ratio	8.71	4.09	0.2	4	6636	-0.6	33.7	116	70	--
		13.5" bore x 16.5" stroke	4° ignition retard	8.66	3.39	0.2	4	6583	-1.4	35.2	--	--	--
		2361 in ³ /cyl	dec. intake manifold air temp. by 30°F	7.98	3.67	0.2	4	6664	-0.2	34.8	--	--	--
		358 hp/cyl	Above plus leaner air-to-fuel ratio using 6% excess air										
		Cooper-Bessemer Company	Above plus water induction at 0.21 water-to-fuel ratio										
		Model KSV-12											
		Ref. 3, 6											

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-6. EMISSIONS SOURCE TEST DATA -- ENGINE 6

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
6	Dual fuel	200 bmeq	Uncontrolled	8.96	4.5	5.16	--	6340	--	--	100	100	28.77
		4300 hp	Derate power 12%	9.56	2.97	6.20	--	7025	10.8	--	83	100	28.77
		600 rpm	4° ignition retard	8.41	7.21	3.22	--	6410	1.1	--	88	100	28.77
		4 cycle	Lean air-to-fuel ratio with 10% excess air	6.70	2.00	7.82	--	6393	0.8	--	86	100	28.77
		Turbocharged	Dec. intake manifold air temp. by 30°F	7.27	4.25	5.28	--	6377	0.6	--	86	70	28.77
		Aftercooled (to 130°F)	Induct water at 0.1 water-to-fuel rate	8.35	3.39	6.48	--	6374	0.5	--	94	100	28.77
		12 cylinder	4° ignition retard & dec. intake manifold air temp. by 30°F	6.63	6.45	3.2	--	6444	1.6	--	88	70	28.77
		13.5" bore x 16.5" stroke	Above plus inc. air-to-fuel ratio to 10% excess air	5.29	3.44	7.39	--	6530	3.0	--	81	70	28.77
		2362 in ³ /cyl	Above plus water induction at water-to-fuel ratio of 0.1	5.27	3.17	7.76	--	6568	3.6	--	--	--	28.77
		358 hp/cyl											
		Cooper-Bessemer Company											
		Model KSV-12											
		Ref. 3, 6											

^aTotal hydrocarbons^bBrake specific fuel consumption^cOperated with minimum pilot oil recommended by manufacturer

TABLE C-7. EMISSIONS SOURCE TEST DATA -- ENGINE 7

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				HC ^a	CO	HC ^a							
7	Dual ^c fuel	Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled Derate power 18%	12.7	1.35	0.85/0.2	--	6400	--	33	20	38	29.03
				10.55	1.7	1.05	--	6600	3.1	32	27	44	29.02

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cOperated with minimum pilot oil recommended by manufacturer

TABLE C-8. EMISSIONS SOURCE TEST DATA -- ENGINE 8

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, Inch Hg
				NO _x	CO	HC _T ^a							
8	Diesel	Repeat from Engine #7	Uncontrolled 4° ignition retard	10.8	1.71	1.0	6	6643	--	35	16	68	29.09
				9.1	--	--	7.5	6881	3.6	35	16	68	29.09

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cSmoke measured with Bosch meter

TABLE C-9. EMISSIONS SOURCE TEST DATA -- ENGINE 9

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	b BSFC, Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				HC _T ^a	CO	HC _T ^a							
9	Diesel	Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled 4° ignition retard Reduce compression ratio	11.4	--	0.44	6	6643	--	--	--	--	--
				9.4	--	0.44	7.5	6817	2.6	--	--	--	--
				10.6	--	--	6.3	6936	4.4	--	--	--	--

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cSmoke measured with Bosch meter

TABLE C-10. EMISSIONS SOURCE TEST DATA -- ENGINE 10

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	b BSFC, Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, in. Hg
				HC ^a	CO	HC _f ^a							
10	Diesel	Large, medium-speed, 2-cycle, blower scavenged engine. Greater than 350 in ³ /cyl	Uncontrolled Derate power 25%	18.8	0.63	0.55	8	7844	--	40	38	68	28.93
				14.5	0.76	0.28	4	8046	2.6	51	38	69	28.93

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cSmoke measured with Bosch meter

TABLE C-11. EMISSIONS SOURCE TEST DATA -- ENGINE 11

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
11	Diesel	Large, medium-speed, 2-cycle, blower scavenged engine. Greater than 350 in ³ /cyl	Uncontrolled Derate power 5%	16.3	0.53	0.18	7.5	6951	--	45.8	47	80	29.02
				12.9	0.64	0.27	3.6	7234	4.1	57.3	50	80	29.02

^aTotal hydrocarbons^bBrake specific fuel consumption^cSmoke measured with Bosch meter

TABLE C-12. EMISSIONS SOURCE TEST DATA -- ENGINE 12

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
12	Diesel	Large, medium speed, 2-cycle turbocharged, aftercooled engine. Greater than 350 in ³ /cy	Uncontrolled 2° ignition retard 4° ignition retard 6° ignition retard 8° ignition retard 10° ignition retard Derate power 25% 5% exhaust gas recirculation 12% exhaust gas	10.7	0.85	0.16	--	6679	--	37	28	70	29.44
				9.8	1.0	0.14	--	6753	1.1	--	--	--	--
				8.6	1.15	0.14	--	6845	2.5	37	31	70	29.43
				7.3	1.2	0.2	10.7	6936	3.8	--	--	--	--
				6.4	1.4	0.14	--	7046	5.5	--	--	--	--
				6.0	1.5	0.14	--	7175	7.4	40	33	70	29.41
				9.9	0.88	0.14	9	--	--	40	22	70	29.34
				9.7	1.2	0.15	10.0	6716	0.5	36	34	70	29.03
				7.1	1.2	0.15	14.8	6716	0.5	33	34	70	29.03

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cSmoke measured with Bosch meter

TABLE C-13. EMISSIONS SOURCE TEST DATA -- ENGINE 13

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				HC ^a	CO	HC ^a							
13	Dual Fuel	Large medium speed, 2 cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled	8.8	1.71	1.54/0.35	--	7040	--	31	40	70	28.92
			2° ignition retard	8.41	1.85	1.42	--	7182	2.0	30	40	70	28.92
			6° ignition retard	6.7	1.98	1.47	--	7284	3.5	31	44	70	28.95
			8° ignition retard	2.6	2.18	1.76	--	7753	10.1	34	42	70	28.95
			Lean air-to-fuel ratio by 17%	6.73	1.86	1.84	--	7220	2.5	36	37	70	28.92
			Dec. manifold air temperature by 20°F (from 120°F)	5.0	0.5	2.4	--	7075	0.5	--	--	70	29.00
			10.7% exhaust gas recirculation	6.6	1.66	1.41	--	7100	0.8	27	56	70	29.02

^aTotal hydrocarbons^bBrake specific fuel consumption^cOperated with minimum pilot oil recommended by manufacturer

TABLE C-14. EMISSIONS SOURCE TEST DATA -- ENGINE 14

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
14	Dual Fuel ^c	Large, medium speed, 2 cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled Inc. speed 25% at constant torque	14.93	1.25	2.66	--	7170	--	36	15	70	29.15
				8.8	1.71	1.54	--	7040	-1.8	31	40	70	28.92

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cOperated with minimum pilot oil recommended by manufacturer

TABLE C-15. EMISSIONS SOURCE TEST DATA -- ENGINE 15

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ₁ ^a							
15	Diesel	150 bmeq	Uncontrolled	11.24	3.02	0.46	Bosch #3	6366	--	31.9	69	73	
		2410 hp	4° ignition retard	10.01	4.44	0.40	Bosch #4	6459	1.5	31.3	52	84	
		450 rpm	Inc. speed to 500 rpm, const. torque	12.07	2.39	0.43	Bosch #3	6250	-1.8	--	58	91	
		4 cycle	Decrease air-to-fuel ratio 12%	11.91	4.65	0.26	--	6522	2.5	28.2	61	68	
		Turbocharged	Derate 27%	10.4	1.56	0.48	Bosch #2	6320	-0.7	36.8	63	--	
		Aftercooled (to 90°F)	Inc. manifold air temp. by 30°F	12.24	1.03	0.42	--	6417	0.8	--	67	72	
		12 cylinder	Dec. manifold air temp. by 20°F	11.54	0.95	0.45	--	6294	-1.1	32.3	66	72	
		13.5" bore x 16.5" stroke	Inc. valve overlap ^c	12.29	1.07	0.45	Bosch #4	6367	0.02	32.0	56	67	
		2361 in ³ /cyl	Induct water at 0.21 water-to-fuel ratio	10.74	2.95	0.51	--	6351	-0.2	31.8	42	70	
		200 hp/cyl	Induct water at 0.58 water-to-fuel ratio	9.6	2.99	0.41	--	6407	0.6	31.7	67	71	
		Cooper-Bessemer Company											
		Model KSV-12-GDT											
		Ref. 8	4° ignition retard & 20°F dec. intake manifold air temp.	9.44	3.84	0.39	--	6514	2.3	31.0	59	87	
			Above plus water induction at 0.21 water-to-fuel rate	8.0	3.62	0.38	--	6496	2.0	31.3	56	75	

^aTotal hydrocarbons^bBrake specific fuel consumption^cDegree of overlap not noted

TABLE C-16. EMISSIONS SOURCE TEST DATA -- ENGINE 16

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, $\frac{\text{Btu}}{\text{hp-hr}}$ ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
16	Dual fuel	150 bmep	Uncontrolled	7.79	3.69	5.67	Bosch #2	6464	--	34.3	44	80	
		2410 hp	2° ignition retard	6.67	--	--	--	6655	2.9	--	45	81	
		450 rpm	3° ignition retard	6.31	4.78	5.08	Bosch #2	6655	2.9	33.7	64	91	
		4 cycle	Inc. air-to-fuel ratio to 10% excess air (at same ratio of pilot oil-to-gas)	5.92	2.02	7.94	Bosch #2	6648	2.8	35.7	61	80	
		Turbocharged											
		Aftercooled (to 90°F)	Inc. rpm to 500 at constant torque	6.42	4.16	5.35	Bosch #3	6571	1.6	--		92	
		12 cylinder	Inc. inlet manifold air temp. by 30°F	9.53	3.24	6.11	--	6516	0.8	--		82	
		13.5" bore x 16.5" stroke	Dec. intake manifold air temp. by 20°F	6.58	3.81	6.34	--	6614	2.3	33.5	46	84	
		2361 in ³ /cyl											
		200 hp/cyl	Induct water at 0.25 water-to-fuel rate	6.90	3.55	6.73	--	6638	2.7	33.0	63	93	
		Cooper-Bessemer Company	Inc. valve overlap by 16°, 3° retard	7.34	4.30	6.07	Bosch #2	6765	4.6	--	56	88	
		Model KSW-12-GDT	3° ignition retard & dec. intake manifold air temp. by 20°F	5.88	8.62	3.03	--	6760	4.6	32.6	47	91	
		Ref. 8	Above plus inc. air-to-fuel ratio by 10% excess air	5.17	4.06	6.38	--	6967	7.8	34.3	47	90	
			Above plus water induction at water-to-fuel ratio of 0.2	4.63	8.35	4.06	--	7255	12.2	33.8	46	84	

^atotal hydrocarbons^bbrake specific fuel consumption

TABLE C-17. EMISSIONS SOURCE TEST DATA -- ENGINE 17

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr ^c			Visible Emission, Percent Opacity	BSFC, Btu/hp-hr ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
17	Diesel	94 bmep	Uncontrolled	14.6	5.4	0.19	--	--					
		2200 hp											
		900 rpm	2° ignition retard (min sac injector)	12.49	6.03	0.18	--	--					
		2 cycle	4° ignition retard (min sac injector)	10.57	6.75	0.19	--	--					
		Blower scavenge (100°F inlet air)	Uncontrolled	18.59	2.56	0.34	--	7478					
		16 cylinder	Increased port width	16.09	2.35	0.34	--	7386					
		9.06" bore x 10" stroke	Uncontrolled	19.09	3.49	0.33	--	--					
		645 in ³ /cyl	Dec. inlet manifold air temp. 25°F (to 100°F)	18.6	1.87	0.35	--	--					
		138 hp/cyl	Uncontrolled	12.56	8.36	0.80	8.5	--					
		Electromotive Division, General Motors Corporation	Install low sac injectors	12.32	7.28	0.64	4.5	--					
		Model 16-645E	Baseline (low sac)	14.96	6.28	0.49	4.5	--					
		Ref. 5	Install minimum sac injectors	14.6	5.4	0.19	--	--					
			Uncontrolled	15.03	5.10	0.49	--	7651					
			Induct water at water-to-fuel ratio of 0.53	12.36	3.71	0.50	--	7560					
			(Continued)										

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cAll data EMD 10 mode composite except retard which is rated load

^dSmoke measured with FHS (light extinction) meter. Also Engines 18, 19, 11-3/4" exhaust stack diameter.

TABLE C-17. Concluded

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr ^c			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
17	Diesel	94 bmepl	Uncontrolled, rated load	15.67	2.53	0.55	--	--	Average of 13 engines Data is EMD composite cycle, 3 variation avg. of 13 engines. Inlet air temperature variation 72° to 107°.				
		2200 hp	Uncontrolled emissions, statistical sample	13.27	0.95	0.35	3	--					
		900 rpm		16.00	2.39	0.62	3	--					
		2 cycle		18.73	3.83	0.89	3	--					
		Blower scavenge (100°F inlet air)											
		16 cylinder											
		9.06" bore x 10" stroke											
		645 in ³ /cyl 138 hp/cyl											
		ElectroMotive Division, General Motors Corporation											
		Model 16-645E											
		Ref. 5											

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-18. EMISSIONS SOURCE TEST DATA -- ENGINE 18

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, $\frac{\text{Btu}}{\text{hp-hr}}$ ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ₁ ^a							
18	Diesel	140 bmep	Uncontrolled ^c	14.74	1.23	0.48	--	6734	-				
		3300 hp	2° ignition retard	12.63	1.43	0.43	--	6826	1.4				
		900 rpm	4° ignition retard	11.36	1.53	0.38	--	6863	1.9				
		2 cycle	Uncontrolled ^c	14.3	1.05	0.36	--	6753	-				
		Turbocharged	Dec. manifold air temp. by 40°F (from 101 to 59°F)	15.3	0.5	0.4	--	6643	-1.6				
		Aftercooled											
		16 cylinder	Uncontrolled	12.70	1.69	1.57	--	--					
		9.06" bore x 10" stroke	Dec. manifold air temp. by 50°F (from 110 to 60°F)	14.98	0.83	0.51	--	--					
		30-1/4" exhaust stack diameter	Uncontrolled (85°F air inlet)	13.85	1.39	0.57	--	--					
		645 in ³ /cyl	Install low sac injectors ^d	14.39	1.42	0.40	--	--					
		206 hp/cyl											
		Electromotive Division, General Motors Corporation	Install experimental min. sac volume injectors (85°F inlet air) ^d	13.79	1.57	0.38	--	--					
		Model 16-645E-3	Uncontrolled, rated load	12.37	1.23	.41	--	--					
		Ref. 5	Uncontrolled emissions, statistical sample	9.97 13.12 16.27	0.92 0.55 2.18	0.16 0.55 0.94	4 4 4	-- -- --					

Data is EMD composite cycle, 30 variation, avg. of 10 engines. Inlet air temp. varied from 72° to 107°F.

^aTotal hydrocarbons^bBrake specific fuel consumption^cRated load^dEMD composite

TABLE C-19. EMISSIONS SOURCE TEST DATA -- ENGINE 19

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, $\frac{\text{Btu}}{\text{hp-hr}}$ ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ₁ ^a							
19	Diesel	133 bmep	Uncontrolled ^c	12.99	1.56	0.59	--	--					
		3900 hp	Install low sac injectors	13.07	1.23	0.30	--	--					
		900 rpm	Uncontrolled ^d , rated load	11.87	0.73	0.37	--	--					
		2 cycle											
		Turbocharged Aftercooled (100°F) 20 cylinders 9.06" bore x 10" stroke 30-1/4" exhaust stack diameter 645 in ³ /cyl 195 hp/cyl ElectroMotive Division, General Motors Corporation Model 20-645E-3 Ref. 5	Uncontrolled emissions, statistical sample -3σ avg +3σ	9.36	0.51	0.30	4	--	Data is EMD composite cycle. 3σ variation, avg. of 11 engines. Inlet air temp. varied from 67° to 97°F.				

^aTotal hydrocarbons^bBrake specific fuel consumption^cEMD composite^dAverage of 11 engines

TABLE C-20. EMISSIONS SOURCE TEST DATA -- ENGINE 20

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, $\frac{\text{Btu}}{\text{hp-hr}}$ ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
20	Gas	140 bmep	Uncontrolled	17.5	0.60	1.2	--	7300	--	--	--	100	28.90
		2400 hp	3° ignition retard	15.24	0.62	1.19	--	7318	0.2	--	--	100	28.90
		330 rpm	5° ignition retard	13.8	0.65	1.2	--	7373	1.0	--	--	100	28.90
		4 cycle	6° ignition retard	13.1	0.65	1.19	--	7391	1.2	--	--	100	28.90
		Turbocharged	9° ignition retard	11.25	0.70	1.19	--	7526	3.1	--	--	100	28.90
		Aftercooled (to 140°F)	10° ignition retard	10.2	0.71	1.19	--	7592	4.0	--	--	100	28.90
		12 cylinder	Uncontrolled	18.1	0.58	1.2	--	7300	0.0	--	--	99	28.96
		16.25" bore x 18" stroke	Dec. intake manifold air temp. 40°F (to 100°F)	12.0	1.0	1.7	--	7264	-0.5	--	--	99	28.96
		3753 in ³ /cyl	Induct water at 0.92 water-to-fuel ratio	7.5	1.5	2.2	--	7375	1.0	--	--	99	28.96
		200 hp/cyl	Dec. speed to 280 rpm at constant torque	21.5	0.41	1.3	--	7337	0.5	--	--	99	28.96
		Ingersoll-Rand Company	Derate power 15%	14.5	0.90	1.28	--	7373	1.0	--	--	100	29.22
		Model 412-KYSR	25%	12.5	1.18	1.60	--	7519	3.0	--	--		
		Ref. 9											

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-21. EMISSIONS SOURCE TEST DATA -- ENGINE 21

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSEC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp, °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
21	Gas	103 bmep	Uncontrolled Retard 50, air-to-fuel change Manifold air temp. decrease of 20°F Manifold air temp. decrease of 20°F and air-to-fuel change Derate 25% Internal EGR Internal EGR plus air-to-fuel charge Internal EGR, air-to-fuel charge and H ₂ O induction	16.6	0.67	--	--	8130	--	17.4	75	90	29.07
		400 hp		11.6	11.5	--	--	8570	5.4	16.8	79	89	29.27
		1000 rpm		16.2	0.81	--	--	8080	-0.6	17.4	30	70	29.27
		4 cycles		12.0	13.1	--	--	8230	1.2	16.9	47	70	29.05
		Naturally aspirated (95°F inlet)		15.5	0.52	--	--	8650	6.4	17.4	82	91	29.03
		6 cylinder		15.7	0.97	--	--	8220	1.1	17.4	68	89	29.17
		8.5" bore x 9 stroke		9.7	7.1	--	--	8470	4.2	16.7	73	90	29.05
		511 in ³ /cyl		3.5	20.9	--	--	9070	11.6	16.5	78	90	29.24
		66.7 hp/cyl											
		White Superior Model 6G-510 Ref. 10, 11											

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-22. EMISSIONS SOURCE TEST DATA -- ENGINE 22

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
22	Gas	107 bmep	Uncontrolled	17.7	1.68	0.8	--	7760	--	18.8	122	103	28.93
		800 hp	Derate 25%, A	11.9	8.27	1.57	--	8507	9.6	17.5	109	103	28.95
		900 rpm	Derate 6%, retard 5°, air-to-fuel change	11.5	7.3 ^c	--	--	8130	4.8	16.1	48	100	29.14
		4 cycle											
		Naturally aspirated (105° F inlet)	Derate 6%, retard 5°, air-to-fuel change, manifold temp. dec. 30°F	14.7	2.7 ^c	--	--	8050	3.7	17.7	24	65	29.14
		8 cylinder	Decrease speed to 750 rpm, constant torque, air-to-fuel change	13.1	5.1	1.21	--	7760	0	17.4	89	96	28.93
		10" bore x 10.5" stroke											
		825 in ³ /cyl											
		100 hp/cyl											
		White Superior Model 86-825 Ref. 10											

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-23. EMISSIONS SOURCE TEST DATA -- ENGINE 23

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, $\frac{\text{Btu}}{\text{hp-hr}}$ ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
23	Gas	147 bmep	Uncontrolled	22.1	0.59	6.87	--	7440	--	26.5	--	102	28.91
		1100 hp	Derate power 25%	24.2	0.52	7.91	--	7550	1.5	27.7	--	102	28.91
		900 rpm	Dec. speed to 750 rpm, constant torque	13.8	6.37	8.6	--	7570	1.7	26.5	--	100	28.58
		4 cycle Turbocharged Aftercooled to 130° 8 cylinder 10" bore x 10-1/2" stroke 825 in ³ /cyl 138 hp/cyl White Superior Model 86T-825 Ref. 10											

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-24. EMISSIONS SOURCE TEST DATA -- ENGINE 24

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
24	Diesel	99 bmep 610 hp 900 rpm 4 cycle Naturally aspirated (95°F inlet) 8 cylinders 9.125" bore x 10.5" stroke 687 in ³ /cyl 76 hp/cyl White Superior Model 40-8 Ref. 10	Uncontrolled Derate power 25% Dec. speed to 750 rpm, constant torque	8.7	8.5	0.25	12	6930	--	26.5	--	90	29.18
				10.7	0.75	0.55	1	6780	-2.2	36.6	--	95	29.16
				8.0	4.5	0.38	6	6550	-5.5	38.4	--	89	29.13

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-25. EMISSIONS SOURCE TEST DATA -- ENGINE 25

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
25	Gas	178 bmep	Uncontrolled Derate 25% Dec. speed to 722 rpm at constant torque	18.7	0.66	--	--	7200	--	25.7	150	89	29.13
		2000 hp		7.9	0.83	--	--	7400	2.8	25.7	174	103	29.12
		900 rpm		24.8	1.21	7.5	--	7820	8.6	20.7	157	90	29.13
		4 cycle											
		Turbocharged											
		Aftercooled 130°F											
		12 cylinders											
		10" bore x 10.5" stroke											
		825 in ³ /cyl											
		167 hp/cyl											
		White Superior											
		Model 12 SGT											
		Ref. 10											

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-26. EMISSIONS SOURCE TEST DATA -- ENGINE 26

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x ^c	CO	HC _f ^a							
26	Diesel	Large, high-speed, 4-cycle, turbocharged engine. Greater than 350 in ³ /cyl.	Uncontrolled 4° ignition retard 8° ignition retard Uncontrolled Dec. intake manifold air temp. by 70°F to 105°F Dec. engine speed to 1000 rpm, constant torque	8.6	2.1	0.4	Bosch #2	7147	--	--	--	--	--
				6.3	2.1	0.3	Bosch #2.4	7421	3.8	--	--	--	--
				5.2	2.1	0.2	Bosch #2.3	7603	6.4	--	--	--	--
				9.3	--	--	--	--	--	--	--	--	--
				7.6	--	--	--	--	--	--	--	--	--
				8.6	--	--	--	--	--	--	--	--	--

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cNO measured only, reported as NO₂

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TABLE C-27. EMISSIONS SOURCE TEST DATA -- ENGINE 27

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, Btu/hp-hr ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x ^c	CO	HC ^a							
27	Diesel	High speed, 4 cycle, turbo-charged engine. Bore greater than 35 in ³ /cyl.	Uncontrolled Inc. exhaust back pressure by 50" of water Inc. exhaust back pressure by 100" of water Derate 25%	7.4	0.8	0.5	Bosch #1.5	6667	--				
				7.25	1.1	0.5	Bosch #2.0	6778	1.7				
				7.1	1.45	0.5	Bosch #2.5	6813	2.2				
				7.1	0.65	0.2	--	--					

^aTotal hydrocarbons^bBrake specific fuel consumption^cNO measured only, reported as NO₂

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TABLE C-28. EMISSIONS SOURCE TEST DATA -- ENGINE 28

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x ^d	CO	HC _T ^a							
28	Gas	Large, high speed, 4 cycle turbo charged engine. Greater than 350 in ³ /cyl.	Uncontrolled	12.5	9.0	0.9	--	8139	--				
			Inc. intake manifold air temp. by 32°F (to 128°F) ^c	8.5	24.0	0.9	--	8139	0				
			Uncontrolled (after cooled to 95°F)	12.0	9.1	0.9	--	7960	--				
			4° ignition retard	11.8	8.5	0.9	--	8020	--				
			6° ignition retard	11.7	8.1	0.9	--	8151	0.2				
			4° ignition advance	11.6	10.5	0.9	--	7929	--				

^aTotal hydrocarbons^bBrake specific fuel consumption^cIncrease in NO_x probably due to change in A/F^dNO measured only, reported as NO₂

TABLE C-29. EMISSIONS SOURCE TEST DATA -- ENGINE 29

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x ^c	CO	HC ^a							
29	Gas	Large, high speed, 4-cycle, turbocharged engine. Greater than 350 in ³ /cyl.	Uncontrolled Manifold air cooling (aftercooled to 85°F) Dec. air-to-fuel ratio by 12% (from 16 to 14.1) Uncontrolled Derate 25%	13.6	18.0	1.0	--	7580	--	16.0			
				13.5	15.0	1.0	--	7590					
				2.0	77.0	1.5	--	8630	13.9	14.1			
				11.85	12.05	1.6	--	--	--				
				11.57	14.70	2.0	--	--	--				

^aTotal hydrocarbons^bBrake specific fuel consumption^cNO measured only, reported as NO₂

TABLE C-30. EMISSIONS SOURCE TEST DATA -- ENGINE 30

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, Btu/hp-hr ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
30	Diesel	4 cycle, turbo-charged, medium bore engine Aftercooled to 110°F	Uncontrolled pre-combustion chamber engine	6.6 ^d	1.9	0.6	--	--					
				5.3 ^e	0.9	0.1	--	--					

^aTotal hydrocarbons^bBrake specific fuel consumption^cNO_x corrected to 75 grains humidity as per 38 FR 32138^dBased on California 13 mode composite test results^eRated condition of California 13 mode test cycle

TABLE C-31. EMISSIONS SOURCE TEST DATA -- ENGINE 31

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
31	Diesel	4 cycle, turbo-charged medium-bore engine	Uncontrolled pre-combustion chamber engine	4.6 ^d	4.1	0.6	--	--					
				3.3 ^e	1.5	0.1	--	--					

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cNO_x corrected to 75 grains humidity as per 38 FR 32138

^dBased on California 13 mode composite test results

^eRated condition of California 13 mode test cycle

TABLE C-32. EMISSIONS SOURCE TEST DATA -- ENGINE 32

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
32	Diesel	4 cycle turbo-charged, medium bore engine	Uncontrolled precombustion chamber engine ^c	4.0 ^d	1.6	0.3	--	--					
				2.7 ^e	1.0	0.1	--	--					

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cNO_x corrected to 75 grains humidity as per 38 FR 32138

^dBased on California 13 mode composite test results

^eRated condition of California 13 mode test cycle

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TABLE C-33. EMISSIONS SOURCE TEST DATA -- ENGINE 33

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b $\frac{\text{Btu}}{\text{hp-hr}}$	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
33	Diesel	4 cycle turbo-charged, medium bore engine	Uncontrolled precombustion chamber engine ^c	5.5	1.3	0.2							

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cBased on California 13 mode composite test results using intermittent rating as 100% power.

TABLE C-34. EMISSIONS SOURCE TEST DATA -- ENGINE 34

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC, ^a							
34	Diesel	1410 hp 1500 rpm 4 cycle Turbocharged 8 cylinder 7.87" bore S.E.M.T. Pielstick Model 8 PA 4200 Ref. 14	Variable throat precombustion chamber ^c Derate 34% with constant rpm	3.5	50 ppm	100 ppm	Bosch #0.96	6869					
				3.5	--	--	Bosch #0.90	6778					

^aTotal hydrocarbons^bBrake specific fuel consumption^cEngine operated at rated condition. Emissions measured with procedure equivalent to DEMA test practice

TABLE C-35. EMISSIONS SOURCE TEST DATA -- ENGINE 35

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
35	Diesel	Less than 500 hp, higher speed, 4-cycle, naturally aspirated engine. Greater than 350 in ³ /cyl.	Uncontrolled	10.0	4.3	0.3	--	7325					

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-36. EMISSIONS SOURCE TEST DATA -- ENGINE 36

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
36	Diesel	Large, high-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl.	Uncontrolled	8.2	1.0	0.3	--	6778					

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-37. EMISSIONS SOURCE TEST DATA -- ENGINE 37

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, $\frac{\text{Btu}}{\text{hp-hr}}$ ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HCT ^a							
37	Gas	Large, high-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl.	Uncontrolled	12.5	7.0	1.5	--	7500					

^aTotal hydrocarbons

^bBrake specific fuel consumption

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TABLE C-38. EMISSIONS SOURCE TEST DATA -- ENGINE 38

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
38	Gas	Large, high-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl.	Uncontrolled 1200 rpm	12.5	11.4	1.5	--	8198					

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-39. EMISSIONS SOURCE TEST DATA -- ENGINE 39

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b $\frac{\text{Btu}}{\text{hp-hr}}$	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
39	Gas	Large, high speed, 4-cycle naturally aspirated engine (Greater than 350 in ³ /cyl.	Uncontrolled	14.0	16.0	1.0	--	9000					

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-40. EMISSIONS SOURCE TEST DATA -- ENGINE 40

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
40	Gas	Large, high-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl.	Uncontrolled	7.8	29.0	1.0	--	8198					

^aTotal hydrocarbons
^bBrake specific fuel consumption

TABLE C-41. EMISSIONS SOURCE TEST DATA -- ENGINE 41

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Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
41	Diesel	141 bmep 1650 hp 900 rpm 2 cycle Turbocharged Aftercooled 8 cylinders 9.06" bore x 10" stroke 645 in ³ /cyl 206 hp/cyl ElectroMotive Division, General Motors Corporation Model 8-645E-3 Ref. 5	Uncontrolled	12.88	0.93	0.42	--	--					

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-42. EMISSIONS SOURCE TEST DATA -- ENGINE 42

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
42	Diesel	94 bmep	Uncontrolled ^c	14.74	4.14	0.49	--	--					
		1650 hp	Uncontrolled ^d	15.52	3.65	0.57	--	--					
		900 rpm											
		2 cycle Blower scavenged 12 cylinders 9.06" bore x 10" stroke 645 in ³ /cyl 138 hp/cyl ElectroMotive Division, General Motors Corporation Model 12-645E Ref. 5	Standard deviation	0.79	0.49	0.09	--	--					

^aTotal hydrocarbons

^bBrake specific fuel consumption

^cAverage of 8 engines (rated load)

^dData is EMD 10 mode composite cycle, standard deviation, σ, average of 8 engines

TABLE C-43. EMISSIONS SOURCE TEST DATA -- ENGINE 43

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
43	Diesel	1100 hp 900 rpm 2 cycle Blower scavenged 8 cylinders 9.06" bore x 10" stroke 645 in ³ /cyl 138 hp/cyl ElectroMotive Division, General Motors Corporation Model 8-645E Ref. 5	Uncontrolled	17.07	2.20	0.37	--	--	--	--	--	--	--

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-44. EMISSIONS SOURCE TEST DATA -- ENGINE 44

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	Δ Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp. °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ₁ ^a							
44	Gas	80 bmeq	Uncontrolled	29.0	0.25	0.30	--	7840	0	21.5	--	--	--
		1650 hp	Increase A/F 2%	23.0	0.25	0.30	--	7840	0	22.0	--	--	--
		330 rpm	Decrease A/F 21%	20.5	1.65	0.35	--	8160	4.1	17.0	--	--	--
		4-cycle, carbureted	Retard 5°	28.2	0.25	0.15	--	8600	9.7	20.0	--	--	--
		Naturally aspirated	Retard 9°	27.9	0.25	0.08	--	9040	15.3	19.0	--	--	--
		12 cylinder	Retard 5° & dec. A/F	14.0	8.4	0.35	--	8720	11.2	16.6	--	--	--
		16.25" bore x 18" stroke	Inc. speed to 380 rpm	25.0	0.30	0.25	--	8240	5.1	21.8	--	--	--
		3733 in ³ /cyl	Derate 25%	6.0	0.30	0.50	--	8160	4.1	24.0	--	--	--
		137.5 hp/cyl											
		Ingersoll-Rand Model PVGR-12 Ref. 16											

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-45. EMISSIONS SOURCE TEST DATA -- ENGINE 45

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HCT ^a							
45	Gas	133 bmep	Uncontrolled	20.0	0.9	1.7	--	7500	--	--	--	--	--
		1500 hp	Retard 5°	15.0	0.9	1.7	--	7540	0.5	--	--	--	--
		550 rpm	Retard 10°	10.0	0.9	1.7	--	7760	3.5	--	--	--	--
		4 cycle	Decrease manifold air temperature to 20°	16.0	1.0	1.8	--	7500	0	--	--	--	--
		Turbocharged	Decrease manifold air temperature to 20°	12.5	1.4	2.0	--	7425	-1.0	--	--	--	--
		Aftercooled 140°F	Derate 25%	9.0	1.8	3.1	--	7760	3.5	--	--	--	--
		12 cylinder											
		11.5" bore x 13" stroke											
		1350 in ³ /cyl											
		125 hp/cyl											
		Ingersoll-Rand											
		Model 12 SVS											
		Ref. 16											

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-46. EMISSIONS SOURCE TEST DATA -- ENGINE 46

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
46	Gas	170 bmep	Uncontrolled	16.5	1.2	2.8/ 0.65	--	6500	--	--	--	--	--
		6000 hp											
		350 rpm	Derate 25%	14.5	1.6	3.0	--	6630	2.0	--	--	--	--
		4 cycle	Retard 4°	14.0	1.1	3.1	--	6630	2.0	--	--	--	--
		Turbocharged	Retard 7°	12.0	1.2	3.5	--	6860	5.5	--	--	--	--
		Aftercooled 110°											
		16 cylinders											
		17" bore x 22" stroke											
		4993 in ³ /cyl											
		375 hp/cyl											
		Ingersoll-Rand											
		Model 616 KVR											
		Ref. 16											

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-47. EMISSIONS SOURCE TEST DATA -- ENGINE 47

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HCT ^a							
47	Diesel	225 bmep 7313 hp 450 rpm 4 cycle Turbocharged Aftercooled 100°F 12 cylinder 17" bore x 21" stroke 4767 in ³ /cyl 609 hp/cyl Delaval Model DSRV-16 Ref. 17	Uncontrolled	9.7	--	0.17	--	6485	--	29.5	54	71	29.96

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-48. EMISSIONS SOURCE TEST DATA -- ENGINE 48

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, Inch Hg
				NO _x	CO	HC _T ^a							
48	Gas	218 bmep 3200 hp 630 rpm 4 cycle Turbocharged Aftercooled 100°F 8 cylinders 14" bore x 15" stroke 2309 in ³ /cyl 400 hp/cyl DeLaval Model HVA-8 Ref. 17	Uncontrolled	19.0	--	3.5	--	6782	--	26.3	55	76	29.84

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-49. EMISSIONS SOURCE TEST DATA -- ENGINE 49

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, Inch Hg
				NO _x	CO	HC ^a							
49	Gas	182 bmep 2300 hp 630 rpm 4 cycle Turbocharged Aftercooled 100°F 8 cylinders 13" bore x 15" stroke 1991 in ³ /cyl 288 hp/cyl DeLaval Model HW-8 Ref. 17	Uncontrolled	18.2	--	3.6	--	7021	--	27.3	51	68	29.99

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-50. EMISSIONS SOURCE TEST DATA -- ENGINE 50

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HCT ^a							
50	Dual fuel	225 bmep 8658 hp 400 rpm 4 cycle Turbocharged Aftercooled 125°F 16 cylinders 17" bore x 21" stroke 4767 in ³ /cyl 541 hp/cyl DeLaval Model DGSRV-16 Ref. 17	Uncontrolled	4.4	--	2.1	--	6225	--	27.9	71	82	29.80

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-51. EMISSIONS SOURCE TEST DATA -- ENGINE 51

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HCT ^a							
51	Diesel	225 bmep 4880 hp 450 rpm 4 cycle Turbocharged Aftercooled 125°F 8 cylinders 17" bore x 21" stroke 4767 in ³ /cyl 610 hp/cyl Delaval Model DSR-8 Ref. 17	Uncontrolled	8.5	--	0.31	--	6650	--	29.4	63	65	29.74

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-52. EMISSIONS SOURCE TEST DATA -- ENGINE 52

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, $\frac{\text{Btu}}{\text{hp-hr}}$ ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
52	Diesel	224 bmep 3631 hp 450 rpm 4 cycle Turbocharged Aftercooled 125°F 6 cylinders 17" bore x 21" stroke 4767 in ³ /cyl 605 hp/cyl DeLaval Model DSR-6 Ref. 17	Uncontrolled	9.4	--	0.27	--	6302	--	31.2	63	68	29.74

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-53. EMISSIONS SOURCE TEST DATA -- ENGINE 53

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
53	Gas	187 bmep 5500 hp 630 rpm 4 cycle Turbocharged Aftercooled 100°F 16 cylinders 14" bore x 15" stroke 2309 in ³ /cyl 344 hp/cyl DeLaval Model HVA-16 Ref. 17	Uncontrolled	16.5	--	3.2	--	6714	--	25.9	41	67	30.02

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-54. EMISSIONS SOURCE TEST DATA -- ENGINE 54

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
54	Gas	182 bmep 4000 hp 630 rpm 4 cycle Turbocharged Aftercooled 100°F 12 cylinder 14" bore x 15" stroke 2309 in ³ /cyl 333 hp/cyl DeLaval Model HVA-12 Ref. 17	Uncontrolled	16.1	--	2.5	--	6851	--	27.4	42	69	29.85

^aTotal hydrocarbons

^bBrake specific fuel consumption

959-1

TABLE C-55. EMISSIONS SOURCE TEST DATA -- ENGINE 55

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp. °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
55	Diesel	240 bmep	Uncontrolled	7.3	1.8	0.3	12	6962	--	29	45	90	29.29
		1336 hp	Retard 2°	7.2	2.0	0.2	14	7072	1.6	29	45	90	29.24
		1100 rpm	Manifold air temp. decrease by 76°F	5.4	2.2	0.2	18	6980	0.3	31	45	80	29.29
		4 cycle	Retard 2°, + Manifold Air Temp. dec. by 76°F	5.4	1.9	0.2	19	7035	1.1	33	45	85	29.26
		Turbocharged Aftercooled 176° 6 cyl 9" bore x 10.5" stroke 668 in ³ /cyl 223 hp/cyl White/Alco 6-25JF Ref 18											

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-56. EMISSIONS SOURCE TEST DATA -- ENGINE 56

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
56	Diesel	214 bmep 1300 hp 900 rpm 4 cycle Turbocharged Aftercooled 8 cylinder 9" bore 10.5" stroke 668 in ³ /cyl 163 hp/cyl White/Alco Model 8-251F Ref. 18	Uncontrolled	8.8	0.5	0.1	--	6668	--	--	45	90	--

^aTotal hydrocarbons
^bBrake specific fuel consumption

1-656

TABLE C-57. EMISSIONS SOURCE TEST DATA -- ENGINE 57

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b $\frac{\text{Btu}}{\text{hp-hr}}$	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
57	Diesel	240 bmep 2675 hp 1100 rpm 4 cycle Turbocharged Aftercooled 12 cylinder 9" bore x 10.5" stroke 668 in ³ /cyl 223 hp/cyl White/Alco Model 21-251F Ref. 18	Uncontrolled	8.4	1.3	0.3	23	6943	--	33	45	85	29.34

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-58. EMISSIONS SOURCE TEST DATA -- ENGINE 58

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
58	Diesel	230 bmep	Uncontrolled Injector hole dec.	4.7	1.8	0.2	28	7328	--	33	45	80	29.10
		2800 hp 1200 rpm 4 cycle Turbocharged Aftercooled 12 cylinder 9" bore x 10.5" stroke 668 in ³ /cyl 233 hp/cyl White/Alco Model 12-251F Ref. 18		7.3	1.0	0.3	21	7181	-2.0	--	45	80	29.10

^aTotal hydrocarbons

^bBrake specific fuel consumption

1-556

TABLE C-59. EMISSIONS SOURCE TEST DATA -- ENGINE 59

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
59	Diesel	262 bmep 3900 hp, intermittent rating 1100 rpm 4 cycle Turbocharged Aftercooled 16 cylinders 9" bore x 10.5 stroke 668 in ³ /cyl 244 hp/cyl White/Alco Model 16-251F Ref. 18	Uncontrolled	7.9	1.3	0.5	--	6888	--	--	45	85	--

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-60. EMISSIONS SOURCE TEST DATA -- ENGINE 60

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
60	Diesel	Large, high-speed, 4-cycle turbocharged engine. Greater than 350 in ³ /cyl	Uncontrolled	8.3	1.6	0.3	Bosch 1.4	6870	--	28.6	35	69	29.25
			Decrease A/F	6.7	6.7	0.1	Bosch 3.6	7511	9.3	21.1	35	69	29.25
			Retard 4°	5.8	1.9	0.2	Bosch 2.2	7017	2.1	28.6	50	69	29.08
			Retard 8°	4.7	1.9	0.2	Bosch 2.3	7383	7.5	28.6	47	71	29.09
			Retard 12°	4.0	1.5	0.3	Bosch 2.3	7603	10.7	29.0	40	70	29.27
			Retard 16°	3.2	1.5	0.2	Bosch 2.2	8152	18.7	27.6	32	73	29.04
			Decrease manifold air temp. 110° (to 130°F)	7.6	2.1	0.2	Bosch 2.0	6998	1.9	28.6	49	75	29.17
			Decrease manifold air temp. 30° to 110°F	7.5	1.5	0.2	Bosch 1.7	6925	0.8	28.6	37	70	29.23
			Retard 8° + A/F dec.	4.2	4.5	0.2	Bosch 3.8	8207	19.5	23.5	47	71	29.09
			Dec. Manifold Air temp. 30° + A/F dec.	6.7	3.3	0.2	Bosch 2.6	7053	2.7	26	37	70	29.23
			Retard 16° + A/F dec.	2.8	3.8	0.1	Bosch 3.7	8647	25.9	21.5	32	73	29.04

^a Total hydrocarbons

^b Brake specific fuel consumption

TABLE C-61. EMISSIONS SOURCE TEST DATA -- ENGINE 61

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
61	Dual fuel	152.2 bmep	Uncontrolled Derate 25% Increase speed to 1000 rpm at constant power (137 bmep)	7.3	2.0	1.5	--	6570	--	29.8	122	92	29.07
		711 hp		5.9	3.0	3.9	--	6970	6.1	30.4	164	92	28.81
		900 rpm		5.1	2.7	4.4	--	6960	5.9	30.6	99	92	28.83
		4 cycle											
		Turbocharged											
		Aftercooled to 100°F											
		6 cylinder											
		9.125" bore x 10.5" stroke											
		687 in ³ /cyl											
		119 hp/cyl											
		White Superior											
		Model 40-GDSX-6											
		Ref. 10											

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-62. EMISSIONS SOURCE TEST DATA -- ENGINE 62

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
62	Diesel	152 bmep	Uncontrolled Derate 25% Increase speed to 1000 rpm at constant power (137 bmep)	4.8	3.9	0.18	--	7040	--	36.3	110	92	28.66
		711 hp		6.6	2.0	0.27	--	6880	-2.3	38.4	110	92	28.66
		900 rpm		5.4	3.1	0.13	--	7060	+0.3	37.7	96	92	29.39
		4 cycle											
		Turbocharged											
		Aftercooled to 100°F											
		6 cylinder											
		9.125" bore x 10.5" stroke											
		687 in ³ /cyl											
		119 hp/cyl											
		White Superior											
		40 -x- 6											
		Ref. 10											

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-63. EMISSIONS SOURCE TEST DATA -- ENGINE 63

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
63	Gas	Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	110° Manifold Air Temperature (80° Ambient)										
			Uncontrolled	16.5	--	--	--	6820		--	48	80	--
			Lean A/F and retard 3°	10.0	--	--	--	7090	04.0	--	46	80	--
			Lean A/F, retard 4°, 30°F manifold temp. reduction	4.3	--	--	--	7430	8.9	--	48	70	--
			Reduced CR, lean A/F	13.2	--	--	--	8080	18.5	--	62	80	--
			Reduced CR, lean A/F, retard 5°	8.5	--	--	--	8360	22.6	--	54	80	--
			130°F Manifold Air Temperature (80°F Ambient, Derated 8%)										
			Lean A/F, reduced CR	14.7	--	--	--	8180	19.9	--	83	100	--
			Lean A/F, retard 3°	11.2	--	--	--	8230	20.7	--	83	100	--
			Lean A/F, retard 3°, 50°F manifold temp. reduction	5.3	--	--	--	8370	22.7	--	83	70	--
			Lean A/F ratio	17.0	--	--	--	7010	2.8	--	48	100	--
			Lean A/F ratio and 3° ignition retard	10.3	--	--	--	7200	5.6	--	48	100	--

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-64. EMISSIONS SOURCE TEST DATA -- ENGINE 64

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
64	Gas	Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	110°F Manifold Air Temperature (80°F Ambient)										
			Uncontrolled	11.7	--	--	--	6760	--	--	106	80	--
			Retard 4°	10.7	--	--	--	7077	4.7	--	106	80	--
			30°F manifold reduction	8.1	--	--	--	6900	2.1	--	--	80	--
			Retard 4°, 30°F manifold reduction	7.0	--	--	--	7150	5.8	--	62	80	--
			Derate 10%	9.4	--	--	--	6900	2.1	--	62	80	--
			130°F Manifold Air Temperature (80°F Ambient Derated 8%)										
			Uncontrolled	12.9	--	--	--	6837	1.1	--	63	100	--
			Derate 20%	13.1	--	--	--	6820	0.9	--	64	100	--

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-65. EMISSIONS SOURCE TEST DATA -- ENGINE 65

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
65	Gas	Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	110°F Manifold Air Temperature (80°F Ambient)										
			Uncontrolled	17.6	--	--	--	7265		--	58	80	--
			Retard 3°	8.9	--	--	--	7510	3.4	--	62	80	--
			Derate 20%	10.2	--	--	--	7580	4.3	--	54	80	--
			130°F Manifold Air Temperature (100°F Ambient Derated 8%)										
			Uncontrolled	16.0	--	--	--	7410	2.0	--	57	100	--
			Retard 3°	8.2	--	--	--	7530	3.6	--	--	100	--

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-66. EMISSIONS SOURCE TEST DATA -- ENGINE 66

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
66	Gas	Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cy	110°F Manifold Air Temperature (80°F Ambient)										
			Uncontrolled	14.0	--	--	--	6780		--	64	80	--
			Lean A/F	10.5	--	--	--	6820	0.6	--	68	80	--
			Lean A/F, retard 3°	8.3	--	--	--	7100	4.7	--	68	80	--
			Lean A/F, 30°F manifold reduction	7.3	--	--	--	6895	1.7	--	84	80	--
			Reduced Compression Ratio										
			Lean A/F	12.2	--	--	--	7560	11.5	--	79	80	--
			Lean A/F, retard 4°	8.5	--	--	--	7910	16.7	--	75	80	--
			Lean A/F, 30° manifold reduction	11.1	--	--	--	7775	14.7	--	84	80	--
			130°F Manifold Air Temperature (100°F Ambient, Derate 8%)										
			Uncontrolled	13.7	--	--	--	6850	1.0	--	70	100	--
			Lean A/F	13.2	--	--	--	7550	11.4	--	74	100	--
			Lean A/F, retard 4°	9.4	--	--	--	7740	14.2	--	79	100	--
			Lean A/F, 20°F manifold reduction	10.6	--	--	--	7830	15.5	--	90	80	--

^a Total hydrocarbons^b Brake specific fuel consumption

TABLE C-67. EMISSIONS SOURCE TEST DATA -- ENGINE 67

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				HC _T ^a	CO	HC _T ^a							
67	Gas	Large, low-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	110°F Manifold Air Temperature (80°F Ambient)	--	--	--	--	6670	--	--	51	80	--
			Lean A/F	--	--	--	--	6660	--	--	51	100	--
			130°F Manifold Air Temperature (100°F Ambient Derated 8%)	12.7	--	--	--	6803	2.1	--	73	100	--
			Uncontrolled	10.5	--	--	--	6650	-0.2	--	55	100	--
			Lean A/F	9.8	--	--	--						
			Lean A/F, 30°F manifold reduction										

^a Total hydrocarbons^b Brake specific fuel consumption

TABLE C-68. EMISSIONS SOURCE TEST DATA -- ENGINE 68

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
68	Diesel	Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled 30°F manifold air temperature reduction (to 100°F) Derate 8% Derate 8%, 30°F manifold air temperature reduction	15.5	--	--	--	6618	100.0	--	50	75	--
				14.6	--	--	--	6540	98.8	--	48	75	--
				15.8	--	--	--	6676	100.9	--	44	75	--
				14.9	--	--	--	6540	98.8	--	47	75	--

^a Total hydrocarbons^b Brake specific fuel consumption

TABLE C-69. EMISSIONS SOURCE TEST DATA -- ENGINE 69

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, Btu/hp-hr ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
69	Gas	Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled A/F increase 4.4% Derate 25%, A/F increase	10.3	1.33	2.48/ 2.62	--	6942	--	32.0	36	70	28.59
				8.0	1.86	2.72/ 2.49	--	7301	5.2	33.4	102	70	29.01
				4.4	2.76	4.66	--	7942	14.4	34.6	102	70	29.01

^a Total hydrocarbons

^b Brake specific fuel consumption

TABLE C-70. EMISSIONS SOURCE TEST DATA -- ENGINE 70

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, Inch Hg
				NO _x	CO	HC _T ^a							
70	Gas	Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled EGR 6.5% EGR 7.3% Derate 17%	8.6	1.66	3.1/ 3.72	--	6845	--	37.4	15	70	28.74
				5.8	1.46	3.05/ 3.64	--	6847	0.0	36.1	13	70	28.60
				4.4	2.51	4.1/ 4.7	--	7210	5.3	35.3	17	70	28.77

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-71. EMISSIONS SOURCE TEST DATA -- ENGINE 71

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
71	Gas	Large medium-speed, 2-cycle, blower scavenged engine. Greater than 350 in ³ /cyl	Uncontrolled	11.5	1.4	2.8	--	7400	--	33.4	--	65	29.12
			Derate 20% (constant speed)	4.6	2.2	3.0	--	7700	4.1	33.4	--	65	29.12
			Derate 20% by decreased speed (const. torque)	7.0	1.8	3.6	--	6900	-6.2	37.0	--	65	29.12

^aTotal hydrocarbons

^bBrake specific fuel consumption

999-1

TABLE C-72. EMISSIONS SOURCE TEST DATA -- ENGINE 72

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
72	Diesel	Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled Derate 25% Air-to-fuel increase, manifold air temperature increase	12.6	0.71	0.25	Bosch 0.4	6508	--	36.5	20	70	29.15
				12.7	0.72	0.23	Bosch 0.4	6601	1.4	39.5	22	70	29.14
				9.1	0.65	0.29	Bosch 0.3	6714	3.2	41.8	4	70	29.66

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-73. EMISSIONS SOURCE TEST DATA -- ENGINE 73

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
73	Diesel	Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl.	Uncontrolled CR, IEGR	9.3	0.57	0.22	--	6809	--	41.7	21	70	29.06
				9.6	0.62	0.18	--	6666	-2.1	41.5	19	70	29.34

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-74. EMISSIONS SOURCE TEST DATA -- ENGINE 74

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	Δ Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, Inch Hg
				NO _x	CO	HC ^a							
74	Diesel	Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled Retard 3°	8.5	0.65	0.22	Bosch 0.7	6741	--	--	67	70	29.18
				6.7	0.84	0.25	Bosch 0.6	6907	2.5	00	62	70	29.17

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-75. EMISSIONS SOURCE TEST DATA -- ENGINE 75

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC ^a							
75	Diesel	Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled A/F change	6.2	3.9	0.4	Bosch 1.2	7035	--	30.8	--	80	29.18
				9.4	1.4	0.4	Bosch 0.6	6668	--	32.8	52	80	29.48

^aTotal hydrocarbons^bBrake specific fuel consumption

959-1

TABLE C-76. EMISSIONS SOURCE TEST DATA -- ENGINE 76

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
76	Dual fuel	Large, low-speed, 4-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled Retard & A/F change	10.6	2.89	1.25	Bosch 0.5	6208	--	35.0	--	80	29.18
				5.4	2.12	3.06	Bosch 0.2	6264	0.9	34.6	64	80	29.58

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-77. EMISSIONS SOURCE TEST DATA -- ENGINE 77

959-1

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, Btu/hp-hr ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
77	Diesel	Large, medium-speed, 2-cycle turbocharged, aftercooled engine. Greater than 350 in ³ /cyl	Uncontrolled Retard 5°, change A/F	11.4	0.91	0.22	Bosch 0.5	6504	--	40.5	37	70	29.07
				7.9	1.08	0.20	Bosch 0.7	6797	4.5	40.8	24	90	29.50

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-78. EMISSIONS SOURCE TEST DATA -- ENGINE 78

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp., °F	Barometric Pressure, Inch Hg
				NO _x	CO	HC ^a							
78	Dual fuel	Large, medium-speed, 2-cycle, turbocharged, aftercooled engine. Greater than 350 in ³ /cyl.	Uncontrolled Retard 2.5°, A/F increase	7.4	1.20	1.00	Bosch 0.05	6297	--	38.2	35	70	29.07
				3.25	1.52	2.17	Bosch 0.05	6495	3.1	42.0	34	70	29.15

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-79. EMISSIONS SOURCE TEST DATA -- ENGINE 79

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp. °F	Barometric Pressure, Inch Hg
				NO _x	CO	HC _T ^a							
79	Gas	Medium-bore, high speed, naturally aspirated engine.	Uncontrolled	18.0	2.2	1.9	--	--	--	--	--	--	--

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-80. EMISSIONS SOURCE TEST DATA -- ENGINE 80

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp. °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
80	Gas	Medium-bore, high speed, naturally aspirated engine.	Uncontrolled	17.2	8.0	1.7	--	--	--	--	--	--	--

^aTotal hydrocarbons^bBrake specific fuel consumption

TABLE C-81. EMISSIONS SOURCE TEST DATA -- ENGINE 81

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, ^b Btu/hp-hr	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp. °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
81	Gas	Medium-bore, high speed, turbocharged, aftercooled engine.	Uncontrolled	16.9	1.4	0.7	--	--	--	--	--	--	--

^aTotal hydrocarbons

^bBrake specific fuel consumption

TABLE C-82. EMISSIONS SOURCE TEST DATA -- ENGINE 82

Engine Code No.	Fuel	Description	Control	Emissions, g/hp-hr			Visible Emission, Percent Opacity	BSFC, $\frac{\text{Btu}}{\text{hp-hr}}$ ^b	% Change in BSFC	A/F	Humidity gr H ₂ O/lb air	Temp. °F	Barometric Pressure, inch Hg
				NO _x	CO	HC _T ^a							
82	Gas	Medium-bore, high speed, turbocharged, aftercooled engine.	Uncontrolled	14.4	1.1	0.8	--	--	--	--	--	--	--

^aTotal hydrocarbons^bBrake specific fuel consumption

C.2 REVIEW OF AMBIENT CORRECTION FACTORS FOR APPLICATION TO LARGE-BORE ENGINE NO_x DATA

Ambient correction factors have been developed for automobile- and truck-size gasoline spark ignition (SI) and compression ignition (CI) engines⁽²³⁾. The following sections will discuss the application of the existing SI factors to natural gas engines (C.2.1) and the existing CI factors to large-bore diesel and dual-fuel engines (C.2.2). In addition, gas turbine ambient correction factors will be examined for application to large-bore engine emissions (C.2.3). As these sections will illustrate, no satisfactory ambient temperature correction factor has been developed for any size of SI internal combustion engines, and only one study has considered ambient temperature corrections for CI engines. Therefore, Section C.2.4 will discuss an analytical approach to correct emissions for variations in ambient temperature.

C.2.1 Ambient Correction Factors Developed for SI Engines

A survey of the literature showed that correction factors for gasoline-fueled engines have been developed only for ambient humidity variations⁽²⁴⁾. One of the studies also evaluated the effect of ambient temperature and barometric pressure variations on exhaust emissions, but found that engine-to-engine variations were too great to generalize a correction factor for either temperature or pressure⁽²⁵⁾. These studies will be briefly discussed in the following paragraphs, and then the ambient humidity correction factors will be compared and evaluated for application to the large-bore natural gas-fueled engines in this study.

The Automobile Manufacturer's Association contracted with the Ethyl Corporation to conduct a study on the effect of ambient air humidity, temperature, and pressure on heavy-duty (HD) gasoline engines⁽²⁶⁾. The

results of this study for the effect of humidity later became the correction factor designated in the Federal Register for HD gasoline engines(27).

This correction factor was derived from emission measurements conducted on seven engines in accordance with the Gasoline-Fueled, Heavy-Duty Engine procedures in the 1970 Federal Register. The test engines were installed in an engine dynamometer test cell where humidity, temperature, and pressure were varied. The engines tested were gasoline-fueled, spark-ignited, carbureted, heavy-duty truck engines ranging in size from 38 to 75 CID per cylinder. The compression ratios varied from 7 to 9.4, and air-to-fuel ratios varied from 14.4 to 15.7. The objective of the study was to develop factors to adjust composite mass emissions to a standard condition of 75 grains H₂O/lb dry air, 90°F inlet air temperature, and a barometric pressure of 29.92 inches of mercury.

Figure C-1(28) illustrates the effect of humidity on both A/F ratio and NO_x emissions (Federal nine-mode composite cycle). As this figure illustrates, a reasonably good correlation was established between changes in ambient humidity and NO_x emissions. Note also that A/F ratios are essentially constant or decrease slightly with increasing inlet air humidity. The composite cycle ambient correction factor was of the form

$$K = 0.634 + 0.00654(H) - 0.0000222(H)^2 \quad (C-1)$$

where $NO_x \text{ corrected} = (K) NO_x \text{ observed}$

H = specific humidity, grains H₂O/lb dry air

Figures C-2(29) and C-3(30) illustrate the effect of temperature and pressure on A/F and NO_x emissions for these same engines. Obviously, engine-to-engine variations were too great to generalize a correction factor for either temperature or barometric pressure.

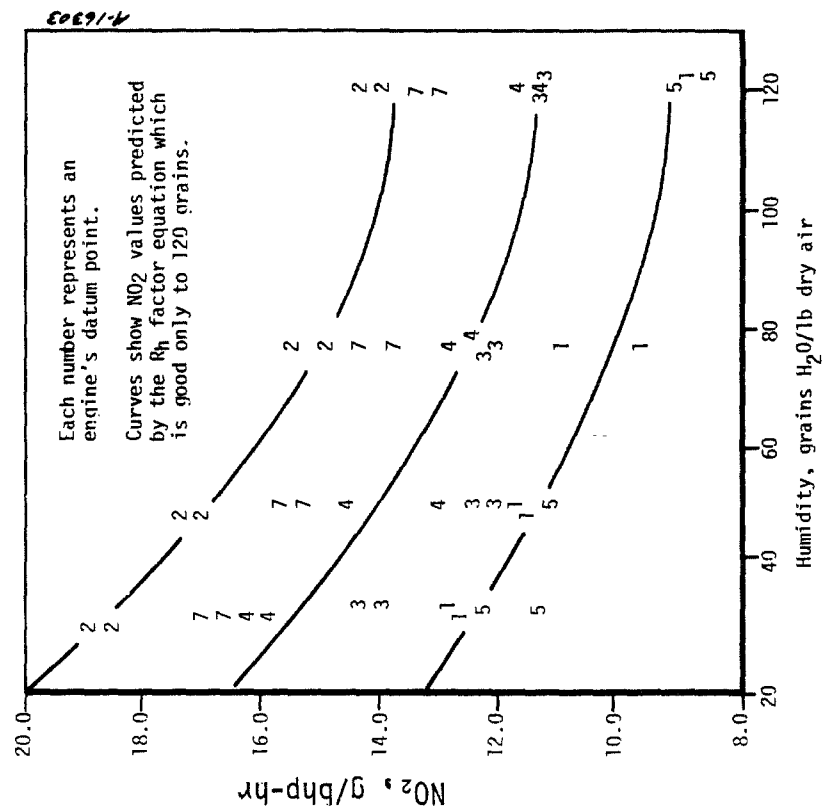
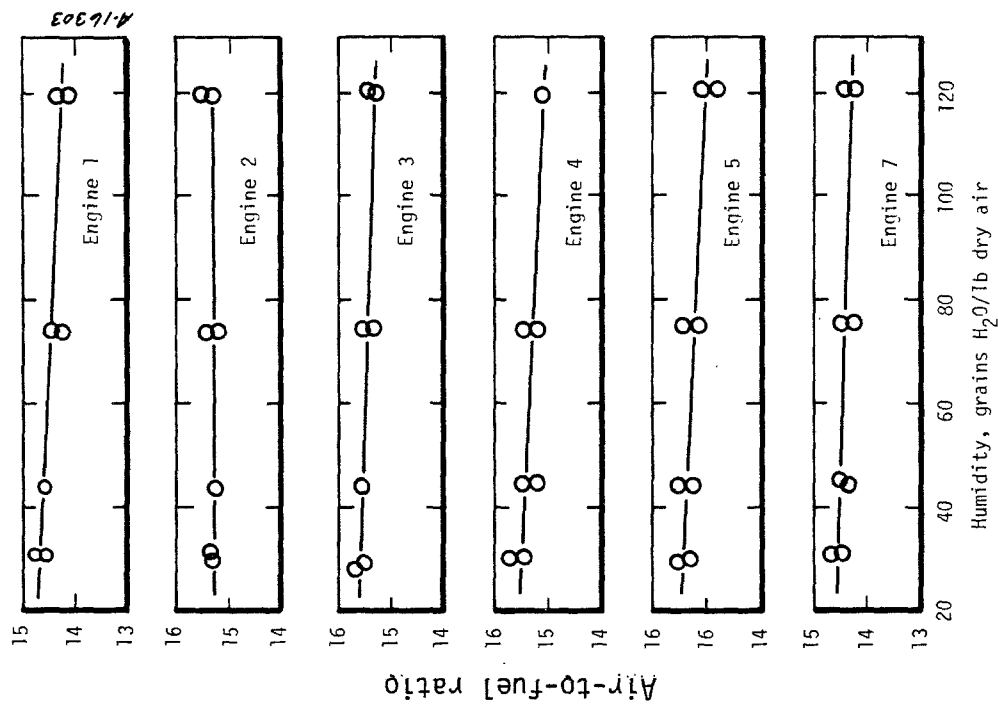


Figure C-1. Effect of humidity on A/F ratio and NO_x emissions for HD gasoline engines (28).

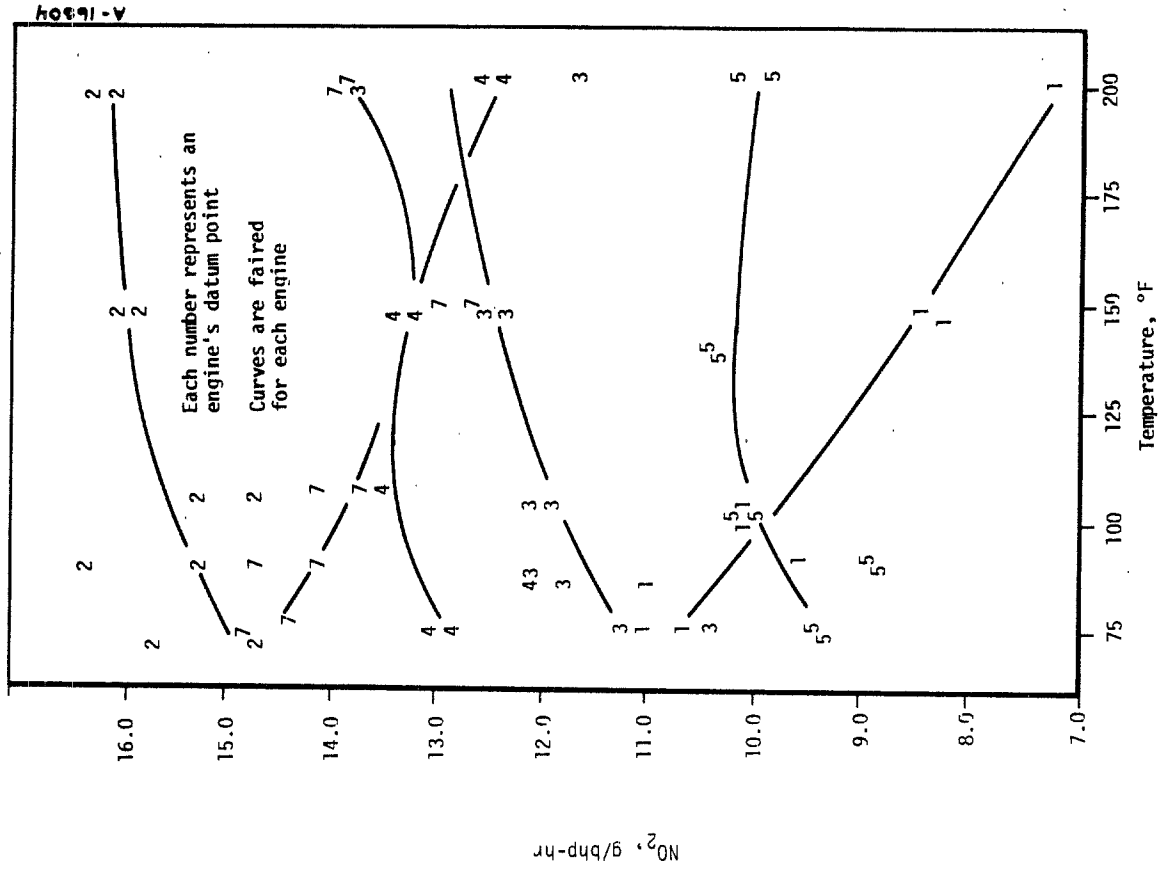
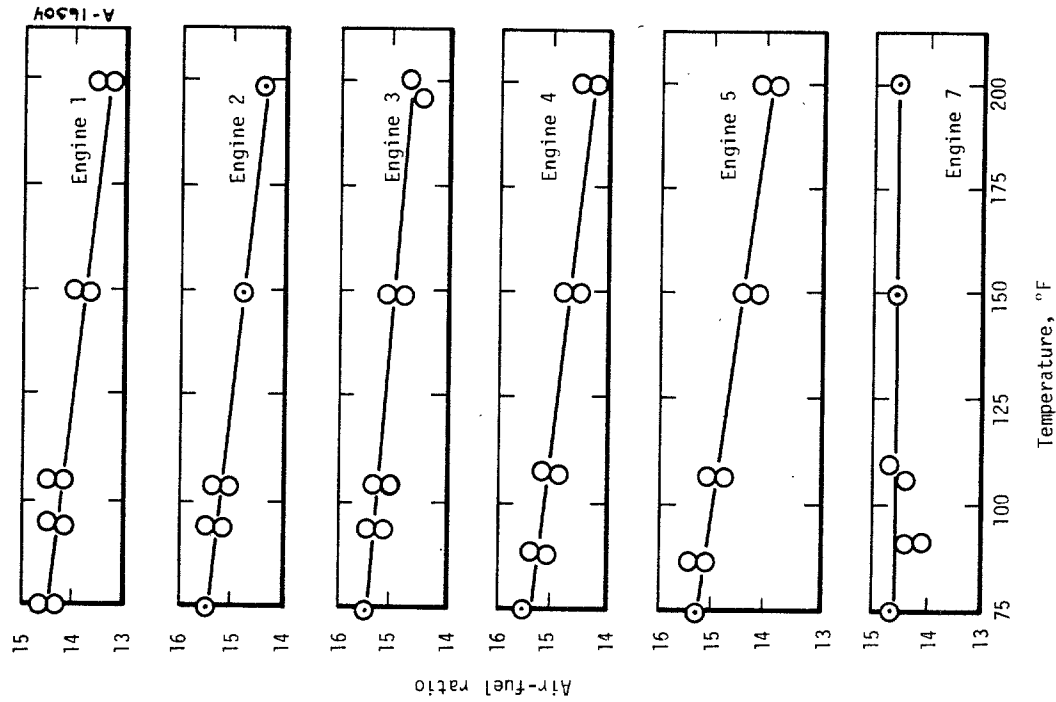


Figure C-2. Effect of temperature on A/F ratio and NO_x emissions for HD gasoline engines (29).

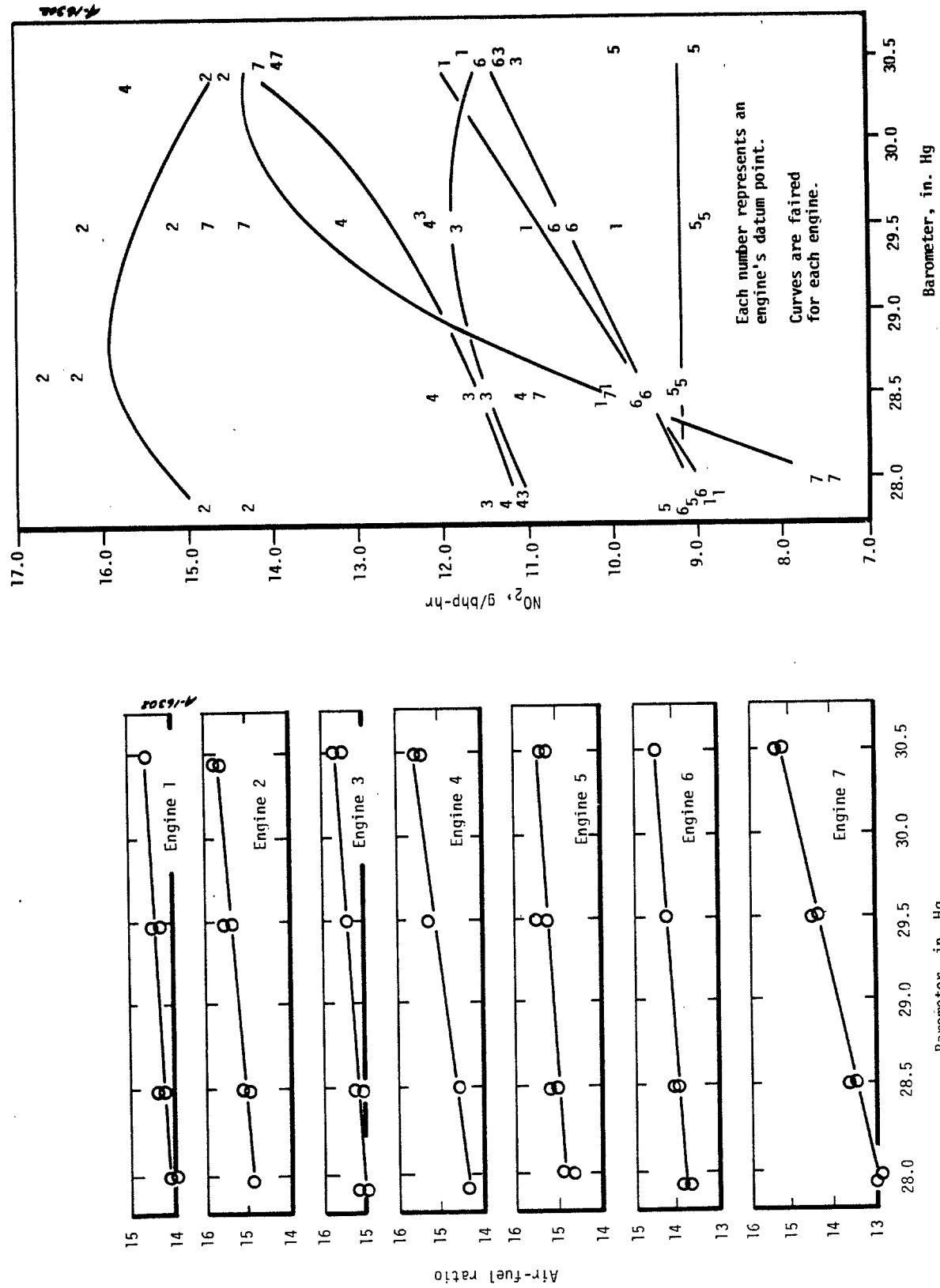


Figure C-3. Effect of barometer pressure on A/F ratio and NO_x emissions for HD gasoline engines (30).

Both physical reasoning and these results indicate that changes in ambient temperature and pressure affect the A/F ratio more than do changes in humidity. Moreover, it is well known that the effect of A/F ratio variations on NO_x emissions depends on the engine operating point on the NO_x vs. A/F curve (see Figure C-4)(31). For example, a decrease in A/F ratio for an engine operating at A in Figure C-4 would result in a decrease in NO_x emissions, whereas a decrease in A/F ratio for an engine at C would cause an increase in NO_x emissions. The author concluded, therefore, that engines operating at different A/F ratios with different metering characteristics should exhibit varying (possibly contradictory) effects on exhaust emissions for similar changes in inlet air conditions.

The author also indicated that changes in inlet air conditions could affect fuel distribution to the engine, and this in turn affects NO_x emissions. This effect is illustrated in Figure C-5(32). In addition, changes in inlet air conditions which change the engine A/F ratio also change the minimum spark advance for best torque (mbt). Figure C-6(33) illustrates how NO_x emissions vary with A/F for the mbt setting and other settings retarded from mbt. Since some of the engines in that study had no vacuum advances, they experienced, in effect, retarded mbt settings as their A/F ratio changed. Thus, their emissions would respond differently to A/F changes than engines with vacuum advances. Therefore, the development of a general correction factor for ambient temperature and pressure was not possible, although variations in these parameters affected emissions significantly.

In two other studies, ambient humidity corrections were developed from LD gasoline vehicles(34,35). In the first study, correction factor was derived from emission tests at 76°F and at four ambient humidities on a fleet of eight passenger cars operated on a chassis dynamometer set to simulate

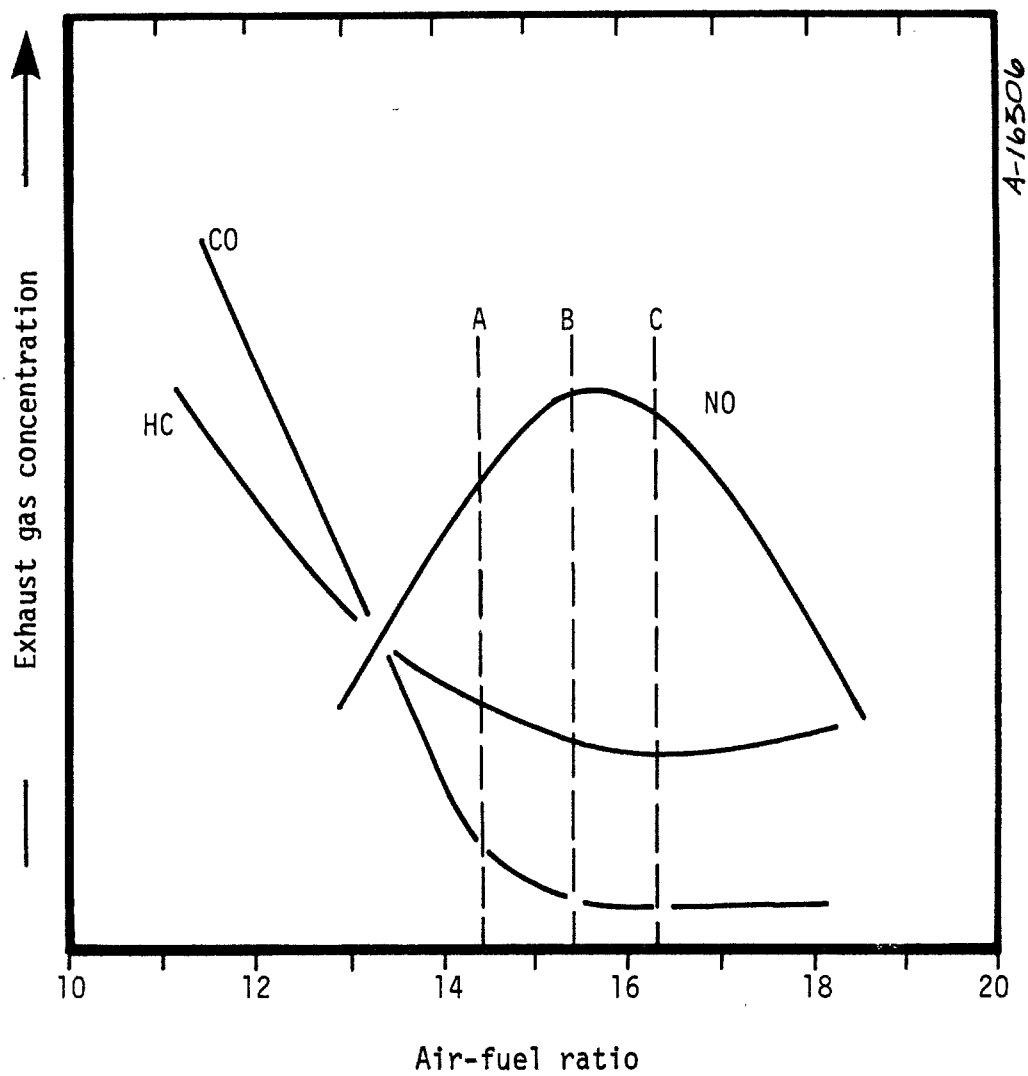


Figure C-4. Exhaust gas concentration versus A/F (31).

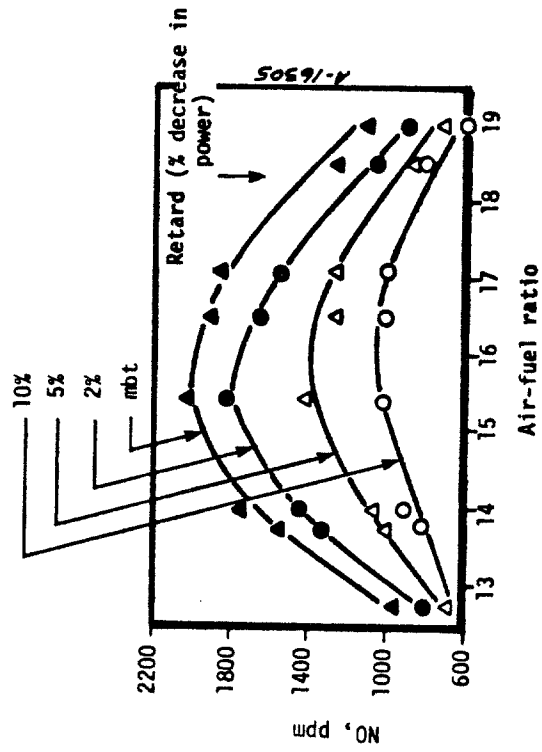


Figure C-5. Effect of fuel distribution on NO emissions (32).

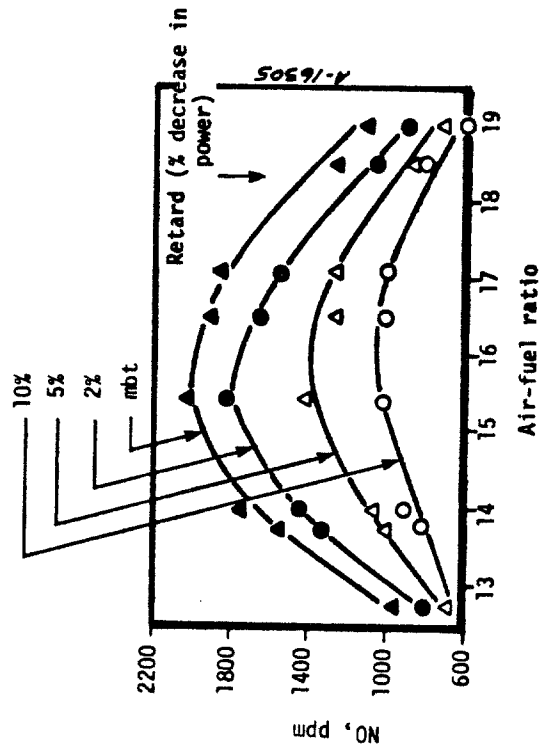


Figure C-6. Effect of ignition timing and A/F on NO exhaust emissions, 1600 rpm, 23 bhp road at mbt (33).

derived from emission tests at 76°F and at four ambient humidities on a fleet of eight passenger cars operated on a chassis dynamometer set to simulate seven different road loads of the Federal seven-mode composite cycle for LD vehicles. These passenger cars were powered by gasoline-fueled, spark-ignited, carbureted engines ranging in size from 42 to 59 CID per cylinder. The compression ratios varied from 8.5 to 10.5, and air-to-fuel ratios from 14.6 to 16.4.

Corrections for ambient humidity were derived for both composite cycle and constant load. The Federal Test Cycle constant load factor is considered more applicable to large-bore carbureted SI engines which typically operate at a constant load (nearly rated load). These factors are

$$\begin{array}{l} K = 0.796 + 0.175(H/100) + 0.129(H/100)^2 \\ \text{composite factor} \end{array} \quad (C-2a)$$

$$\begin{array}{l} K = 0.844 + 0.151(H/100) + 0.075(H/100)^2 \\ \text{constant load, 50 mph} \end{array} \quad (C-2b)$$

The results of this study were adopted by California to correct emissions for ambient humidity for gasoline-powered vehicles under 6000 pounds⁽³⁶⁾.

In the second study, a similar test program on gasoline vehicles was conducted to develop an ambient humidity correction factor that was adopted by EPA to correct LD gasoline vehicles^(37,38). This factor is

$$K = 1/(1 - 0.0047(H - 75)) \quad (C-3)$$

In yet another study, an ambient correction factor for humidity was developed for typical research engines where A/F and mbt settings were held constant⁽³⁹⁾. The author of this study also noted the effect of A/F ratio on NO_x emissions as illustrated in Figure C-7⁽⁴⁰⁾. Based on this figure,

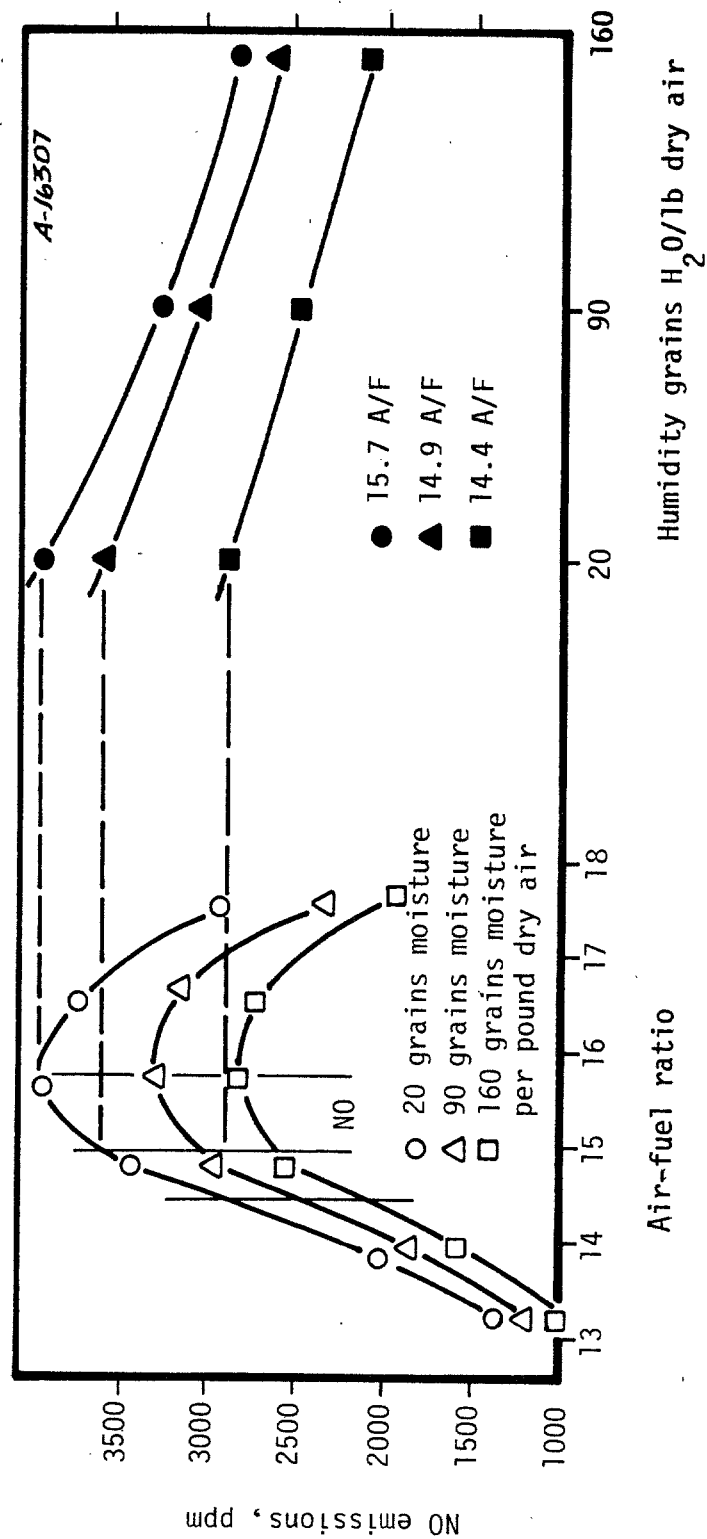


Figure C-7. Effect of A/F ratio and ambient humidity on NO emissions of a LD gasoline engine (40).

Krause⁽⁴¹⁾ reasoned that composite correction factors would vary from constant load factors since NO_x emissions vs. humidity curves changed with A/F ratio. Despite the validity of these observations, the correction factor derived in this study is probably less applicable to large-bore engines than those derived for gasoline vehicles, since in practice A/F ratios and spark settings of large-bore engines are not necessarily held constant as they were in this study (e.g., spark timing is fixed as load changes for some engines, and varies with load for other engines).

Comparison of Existing Humidity Correction Factors for SI Engines

Based on the preceding discussion, three ambient humidity correction factors are potentially applicable to large-bore, natural gas fueled engines, particularly four-stroke, carbureted versions. These factors are summarized in Table C-83^(42,43,44,45). Figure C-8 is a comparison of the three factors over a typical range of ambient humidities. Note that only one of the factors plotted is a constant load factor (Equation (C-2b)); the other three are based on composite test cycles.

As Figure C-8 illustrates, there is a considerable difference in correction depending on the study. The results from the EPA/Scott study show the greatest sensitivity to ambient humidity variation, while the results for HD gasoline engines show the least. This is not unexpected in view of the previous discussion regarding the variation in engine responses to changes in inlet conditions depending on A/F ratio, fuel metering and distribution, and ignition (distribution operation) characteristics of different engines. The correction based on constant load has been chosen as the most suitable correction to be applied to large-bore SI engines, since these engines are

TABLE C-83. AMBIENT HUMIDITY CORRECTION FACTORS FOR SI ENGINES

Equation No.	Correction Factor	
(1)	$K = 0.634 + 0.00654(H) - 0.0000222(H)^2$ Composite Factor (9 Mode Federal HD Gasoline Test Cycle)	(42)
(2a)	$K = 0.796 + 0.175(H/100) + 0.129(H/100)^2$ Composite Factor (Federal Test Cycle, LD Gasoline Vehicles)	(43)
(2b)	$K = 0.844 + 0.151(H/100) + 0.075(H/100)^2$ 50 mph, Constant Load	(44)
(3)	$K = 1/(1 - 0.0047(H-75))$ Composite Factor (Federal Test Cycle, LD Gasoline Vehicles)	(45)

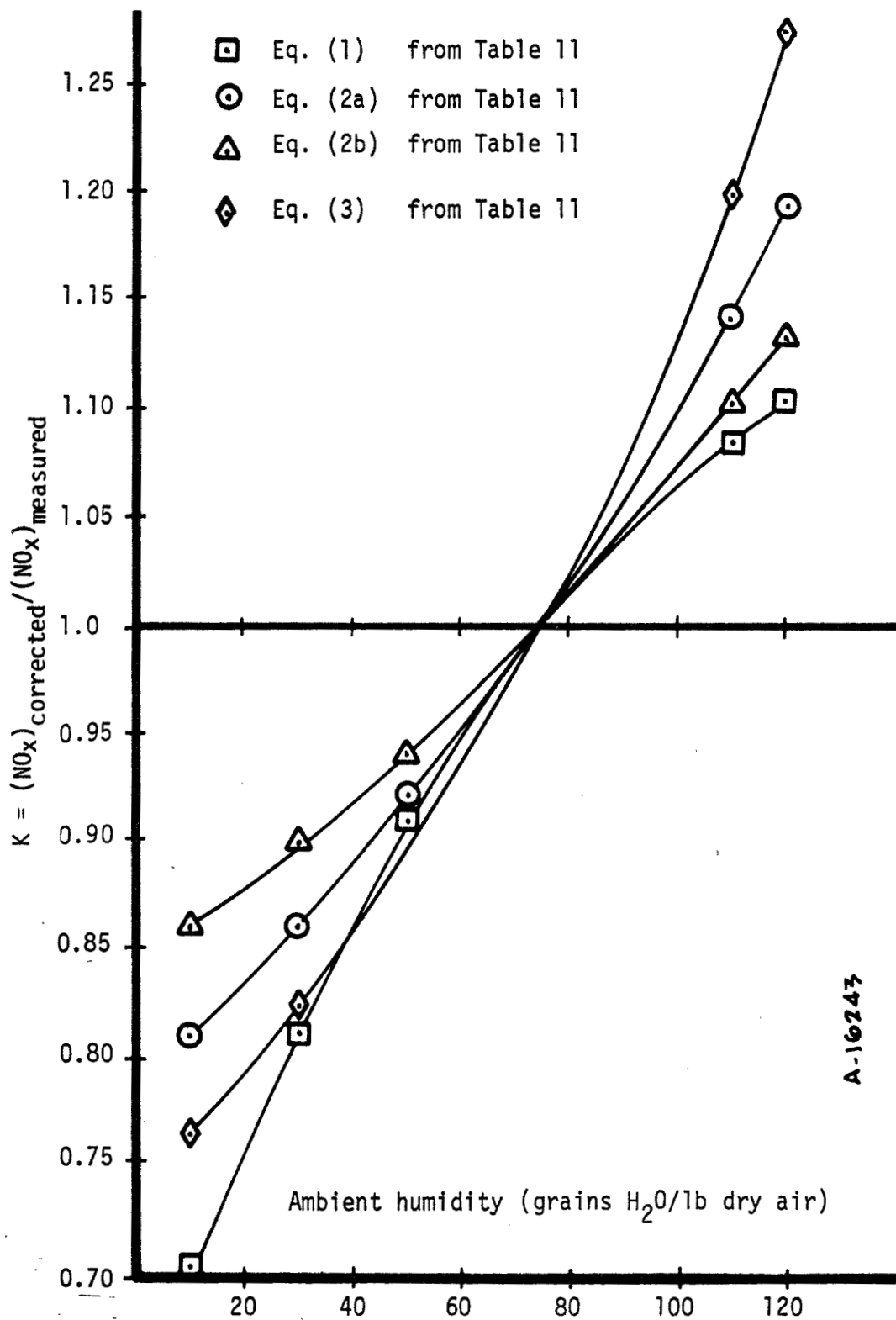


Figure C-8. Comparison of SI ambient humidity correction factors.

typically operated at constant load. Thus, this correction factor is most applicable to carbureted (4-NA) natural gas engines.

Application of any of the correction factors in Figure C-8 to other engine types (e.g., turbocharged units) is questionable due to major differences in inlet air intake systems. For example, data from the draft SSEIS indicate that NO_x reductions due to water induction are directly proportional to water-to-fuel (W/F) rates, up to W/F ratios of 1. Moreover, W/F rates due to ambient humidity are a function of A/F ratio, as Figure C-9 illustrates. Based on this figure, the W/F rate of a carbureted natural gas engine (trapped A/F ≈ 17) is about 30 percent lower for a specific humidity of 100, than the W/F ratio of a turbocharged engine with a trapped A/F ratio of 25 (trapped A/F ratios for turbocharged SI engines typically range from 20 to 25)(46,47). Note, also, that the curves of constant humidity diverge with increasing A/F ratio. Therefore, it can be anticipated that NO_x emissions of engines with different A/F ratios will respond differently to identical changes in ambient humidity. Thus, the application of the constant load humidity correction factor (based on carbureted gasoline engines) to other than carbureted, large-bore SI engines is questionable.

Similar conclusions can also be reasoned regarding the application of ambient temperature and pressure correction factors to NO_x emissions from engines whose air intake as well as fuel systems differ substantially. As discussed earlier, however, no factors have been developed for SI engines for either temperature or pressure.

C.2.2 Ambient Correction Factors Developed for CI Engines

A survey of the literature established that two sources have reported ambient correction factors for truck-size diesel engines(48,49). A study by Krause, et al., was sponsored by the Automobile Manufacturer's Association

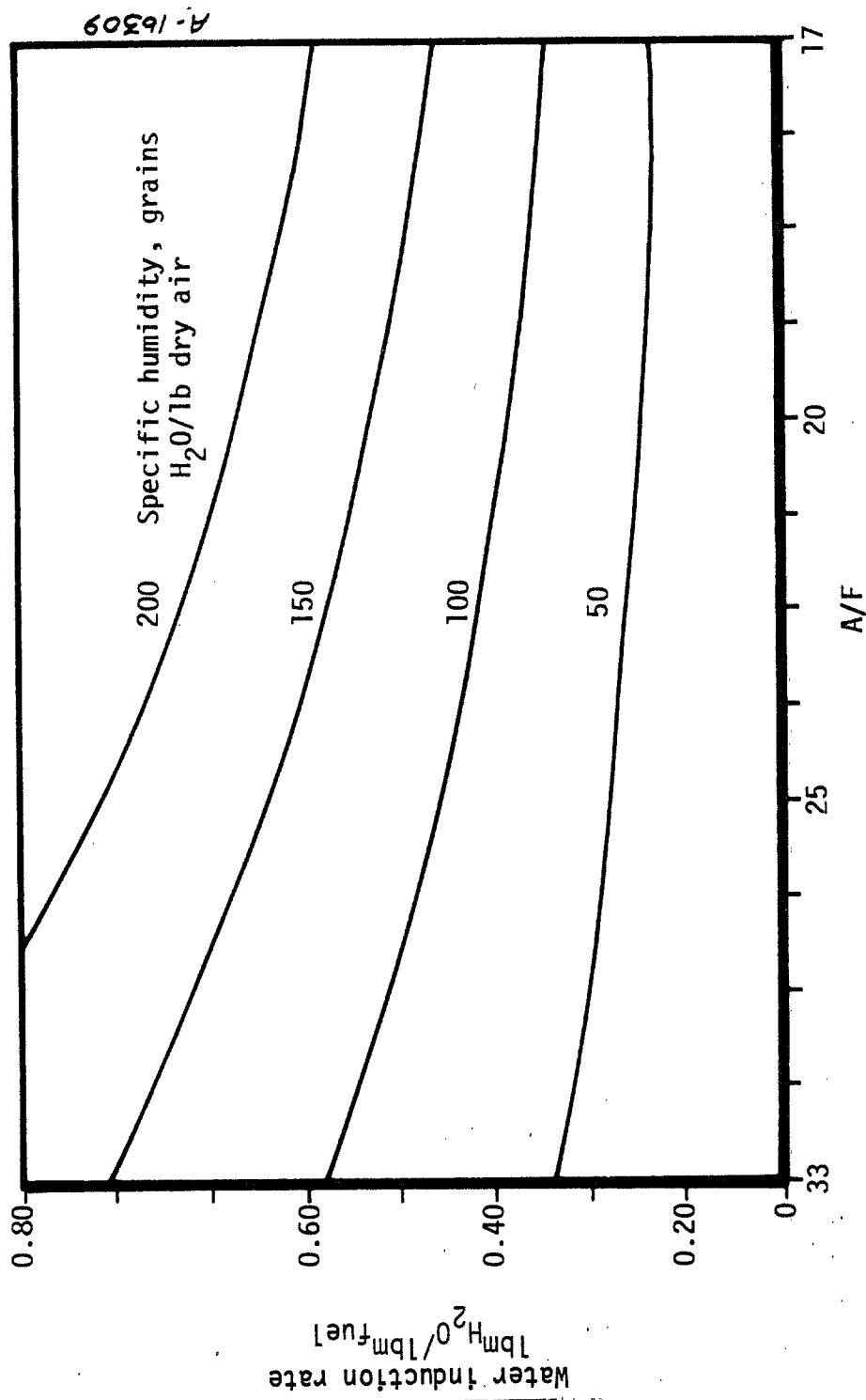


Figure C-9. Water-to-fuel range versus ratio air-to-fuel.

and the Engine Manufacturer's Association and resulted in correction factors for both temperature and humidity. These corrections were later adopted by the EPA for HD diesel engines⁽⁵⁰⁾. The other source reported corrections for humidity only.

In Krause's study, a correction factor was derived for six different engines run over a 13-mode cycle. Emission procedures used were similar to "California Procedures for Diesel Engines in 1973 and Subsequent Model Year Vehicles Over 6001 Pounds Gross Vehicle Weight." The heavy-duty truck engines tested were diesel-fueled, compression-ignition engines ranging in size from 69 to 155 CID. The compression ratios varied between 14.9 and 18.7. Inlet air conditions were controlled to one of 19 combinations of humidity and temperature. The humidity conditions ranged from 35 to 125 grains H₂O/lb dry air, and the temperature was varied between 70°F and 115°F. Barometric pressure was controlled to 28.00 ± 0.2 inch Hg at the air cleaner inlet.

The engines in the study were of the following types:

- Four-stroke turbocharged, direct injection (4-TC)
- Four-stroke turbocharged, prechamber (4-TC, PC)
- Four-stroke naturally aspirated, MAN chamber (4-NA, MAN)
- Four-stroke naturally aspirated, (4-NA)
- Two-stroke blower scavenged (2-BS)
- Four-stroke turbocharged, aftercooled (4-TC, AC)

Unlike turbocharged large-bore engines, which are nearly always aftercooled, only one of the turbocharged engines in this study was aftercooled. That engine as well as the 4-NA and the 2-BS units were similar in design to large-bore engines. A correction factor was developed for all of the engines as well as for individual units. The factor was of the form

$$K = 1/(A(H - 75) + B(T - 85))$$

where NO corrected = $(K)_x$ (NO observed) and A and B are related to A/F ratio (load). The reference ambient conditions are 75 grains H₂O/lb dry air and 85°F. Figure C-10⁽⁵¹⁾ is a plot of the coefficients A and B for each engine as a function of load (or A/F ratio).

Average values of A and B for all engines (as function of A/F) were also determined. This result does not appear wholly justified given the variations in response to ambient humidity and temperature exhibited by the different engine types depicted in Figure C-10. Note that the 4-TC, AC unit is significantly less sensitive to ambient temperature variations over the load range than the other designs. Therefore, separate ambient correction factors for 2-BS, 4-NA, and 4-TC, AC units from this study were used on the corresponding large-bore designs instead of just one average value of all engines. The coefficients A and B were determined for rated load conditions using average rated load A/F ratios that were reported for the large-bore engines.

The second source⁽⁵²⁾ which examined humidity effects reported correction factor based on experimental tests of a 2-BS and a 4-NA engine.

Ambient humidity corrections from both of these sources (assuming inlet temperature is held constant, hence, the B term of Krause's factor drops out) are illustrated in Figure C-11^(53,54). The SI (gasoline) factors discussed in Section C.2.1 are also plotted for comparison. Note that there is little variation in correction factors for different diesel engine types with the exception of the 2-BS unit from the CRC report. Moreover, the diesel-fueled (CI) engines appear less sensitive to ambient humidity changes

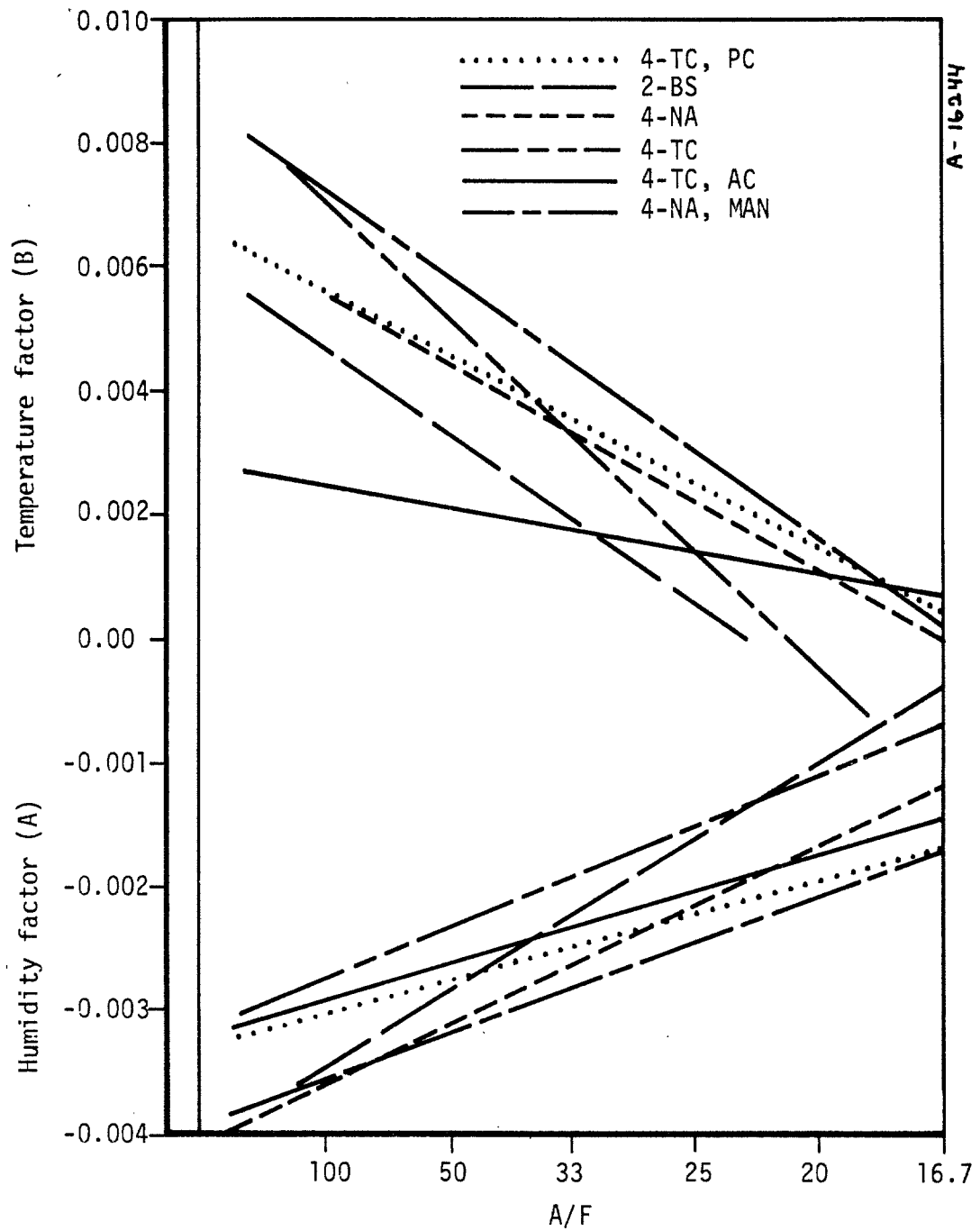


Figure C-10. Effect of humidity and temperature on NO emission for different CI engines (51).

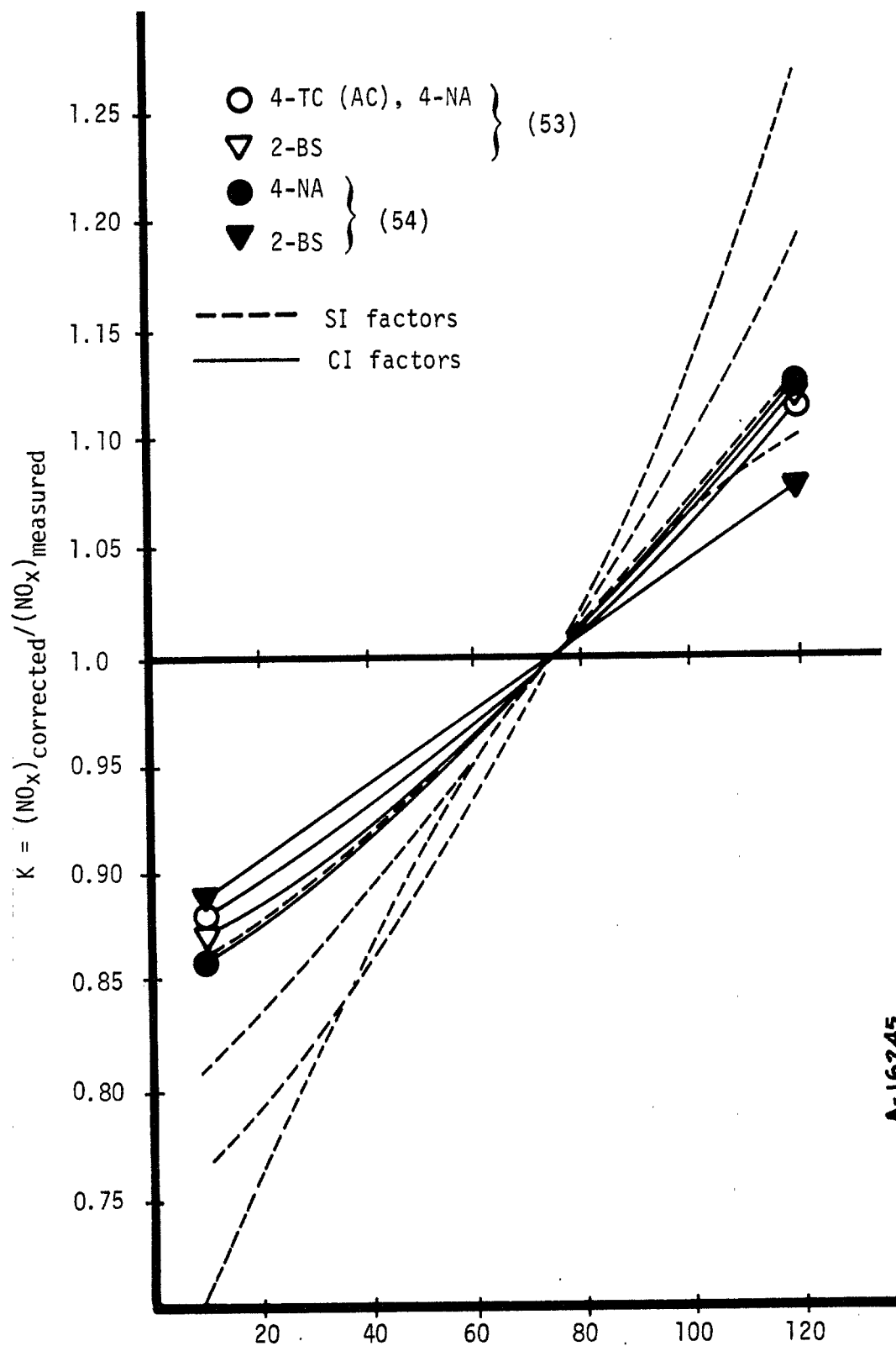


Figure C-11. Comparison of CI and SI ambient humidity correction factors.

than SI units, particularly for humidities less than 75 grains H₂O/lb dry air.

Initially, one might expect the diesel units to be more responsive to inlet humidity variations, since these units operate at higher A/F ratios (approximately 25 to 40), and therefore, induct more ambient water than SI engines which operate more nearly at stoichiometric ratios (\approx 15 to 17, depending on the fuel). Diesel engines, however, have a greater thermal inertia than SI engines due to their higher trapped A/F ratios (as well as a different combustion process). Apparently the higher thermal inertia in the diesel units more than offsets their higher effective water induction rate; thus, their NO_x emissions are less sensitive to changes in ambient humidity than SI engines. This explanation is corroborated by experiments which have demonstrated that water injection, as a control technique, produces significantly greater NO_x reductions in SI units than it does in CI (diesel) units (see Section 4.4.7 of the draft SSEIS).

The Krause study also investigated the effect of ambient temperature on NO_x emissions. Figure C-12⁽⁵⁵⁾ presents the correction factors that were derived for engine types similar to those in the present study (humidity is assumed constant, hence, the A term of Krause's factor drops out). The reference temperature is taken as 85°F. This figure shows that the naturally aspirated and blower-scavenged engines NO_x emissions are more sensitive to inlet air temperature changes than the aftercooled design. Since Krause's study systematically examined the effect of both inlet temperature and humidity for a number of CI engine types, his factors were selected for application to similar large-bore engine types.

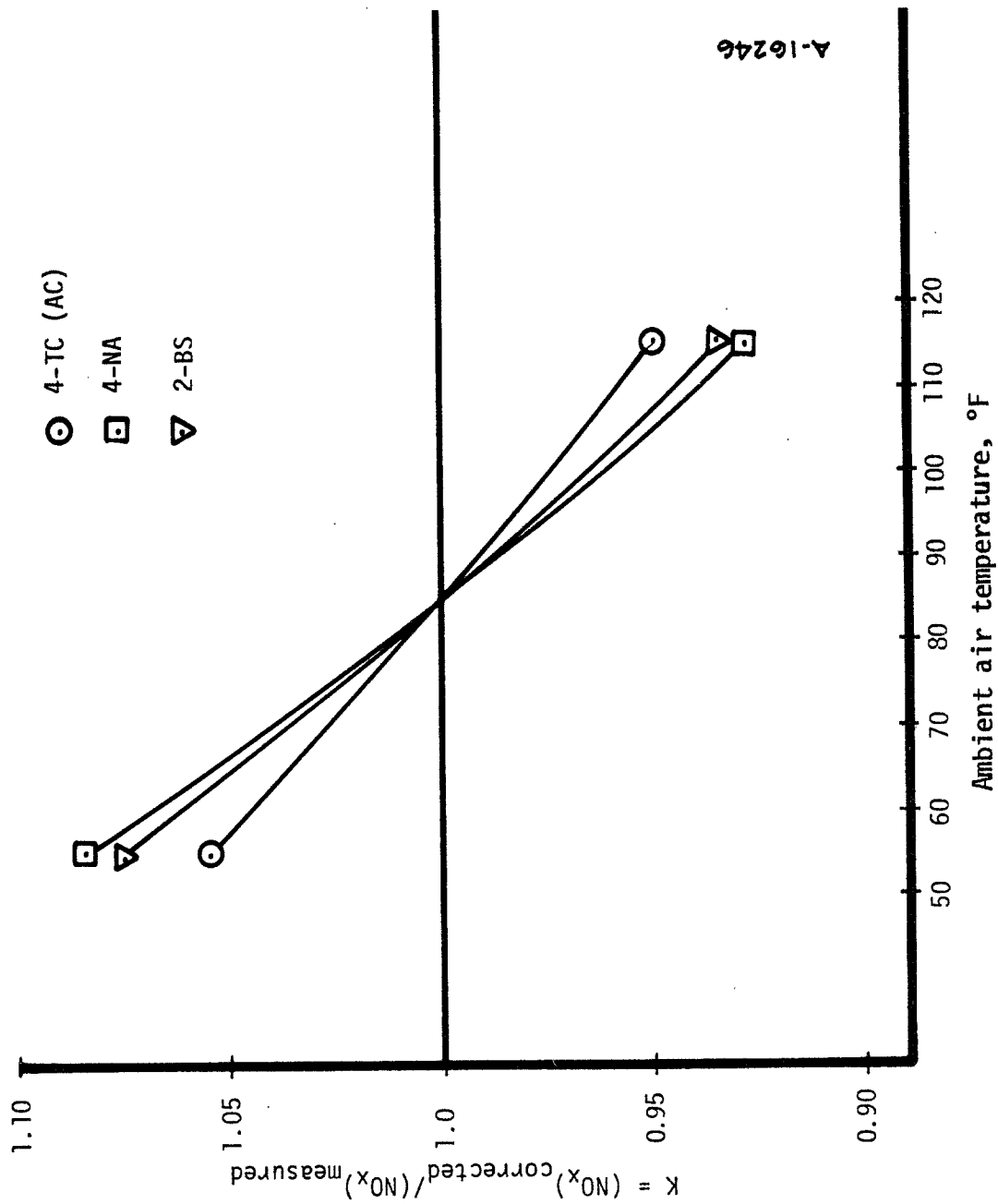


Figure C-12. Correction factors for temperature for CI engines (55).

C.2.3 Potential Application of Gas Turbine Ambient Correction Factors to Reciprocating IC Engines

Since no ambient correction factors have been developed specifically for large-bore engines, all existing correction factors for internal combustion engines were examined including those for gas turbines. There is a considerable amount of technical literature on gas turbine ambient correction factors as a result of the promulgation of emission standards for aircraft engines and the proposed emission standards for stationary gas turbines(56-61). This section describes these various factors and their potential application to the large-bore engine data in the present study.

Gas turbine combustion characteristics have some common features with those in diesel engines. Both the gas turbine primary combustion zone and the diesel combustion chamber can be characterized as a well-stirred reactor, and both use similar distillate fuels or natural gas. On the other hand, the gas turbine combustion is a steady-state, constant pressure process, whereas the diesel is an unsteady, variable pressure combustion process.

Nevertheless, the similarities of the two systems warrant an investigation of existing gas turbines correction factors.

Many investigators have developed ambient humidity correction factors based on a model that relates NO_x formation parameters of temperature, pressure, equivalence ratio and residence time (using the kinetic rate equations for NO_x formation)(62). Humidity enters this model through its effect on reaction (flame) temperature. Most researchers have shown that the effect of humidity on NO_x formation takes the form

$$\text{NO}_x \text{ corrected} / \text{NO}_x \text{ observed} = \exp [K (H_{\text{observed}} - H_{\text{reference}})]$$

where H_{ref} = specific humidity at reference (standard) conditions
K = empirical constant that ranges from 14 to 30, generally
taken as 19⁽⁶³⁾

Figure C-13^(64,65) is a plot of this correction factor and shows that it agrees reasonably well with the HD diesel (4-TC, AC) factor discussed in Section C.2.2 (the gas turbine factor was adjusted to a reference humidity of 75 grains H_2O/lb dry air to correspond with the reference humidity of the HD factor).

Other sources have shown similar agreement of humidity effects on NO_x emissions of diesel engines and gas turbines as illustrated in Figure C-14⁽⁶⁶⁾. The gas turbine humidity correction factor was derived from empirical data based on water injection as a means of NO_x control in gas turbines. The ambient humidity was converted into an effective water-to-fuel ratio by multiplying the ambient humidity loading by the near stoichiometric A/F ratio existing at combustion. Then the empirically derived water injection correction factor of Ambrose⁽⁶⁷⁾ was used to calculate the percentage reduction in NO_x . It is reasoned that the overall A/F ratio is inappropriate since much of the water vapor in the inlet air never reaches the primary combustion zone because it is vented for engine cooling or enters downstream as dilution or wall cooling air. Based on this result, one can conclude that changes in humidity appear to affect NO_x formation in gas turbines in much the same way as diesel (IC) engines. However, it should be remembered that a number of adequate humidity correction factors have been developed for IC engines; therefore, this gas turbine result is of limited value, but serves to reinforce the more global application of these corrections.

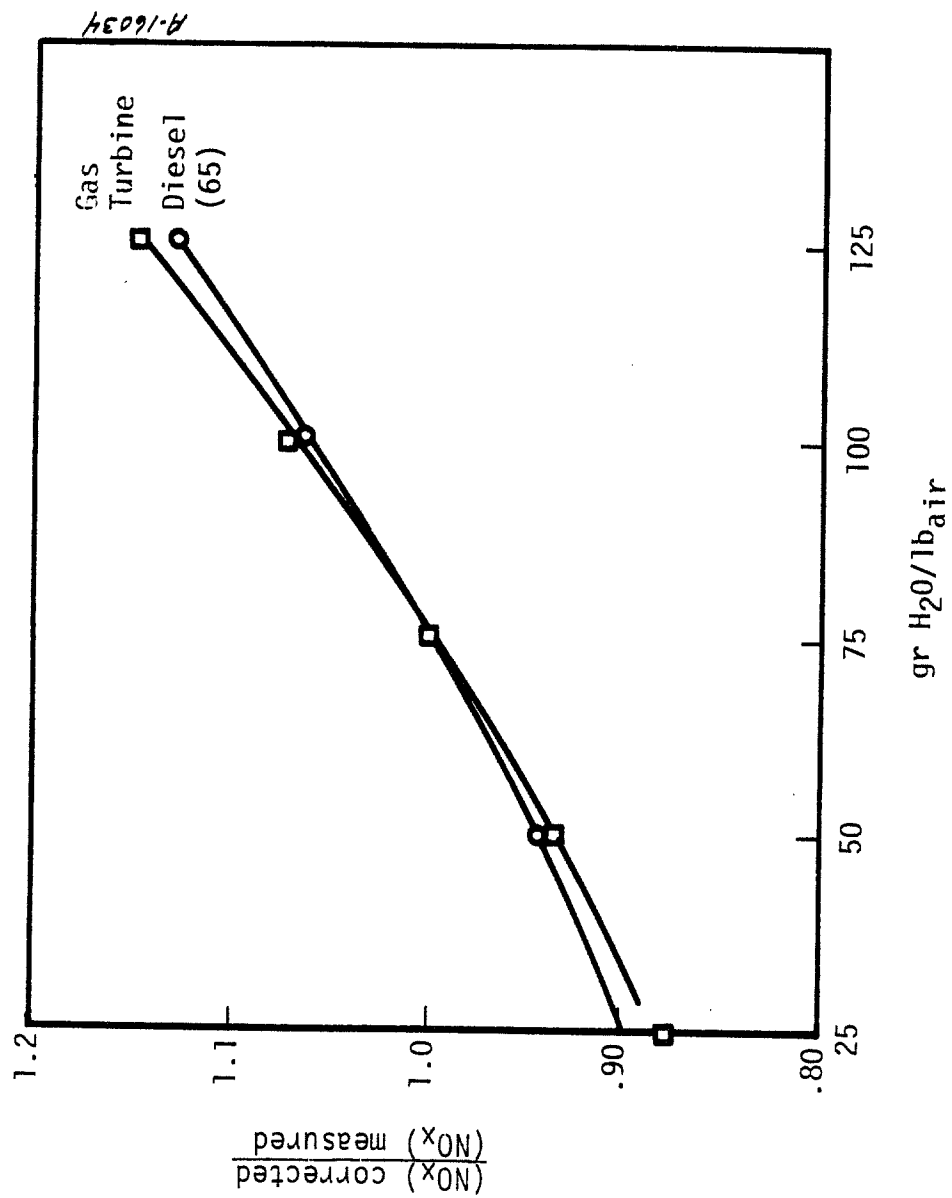


Figure C-13. Gas turbine humidity correction factor (64).

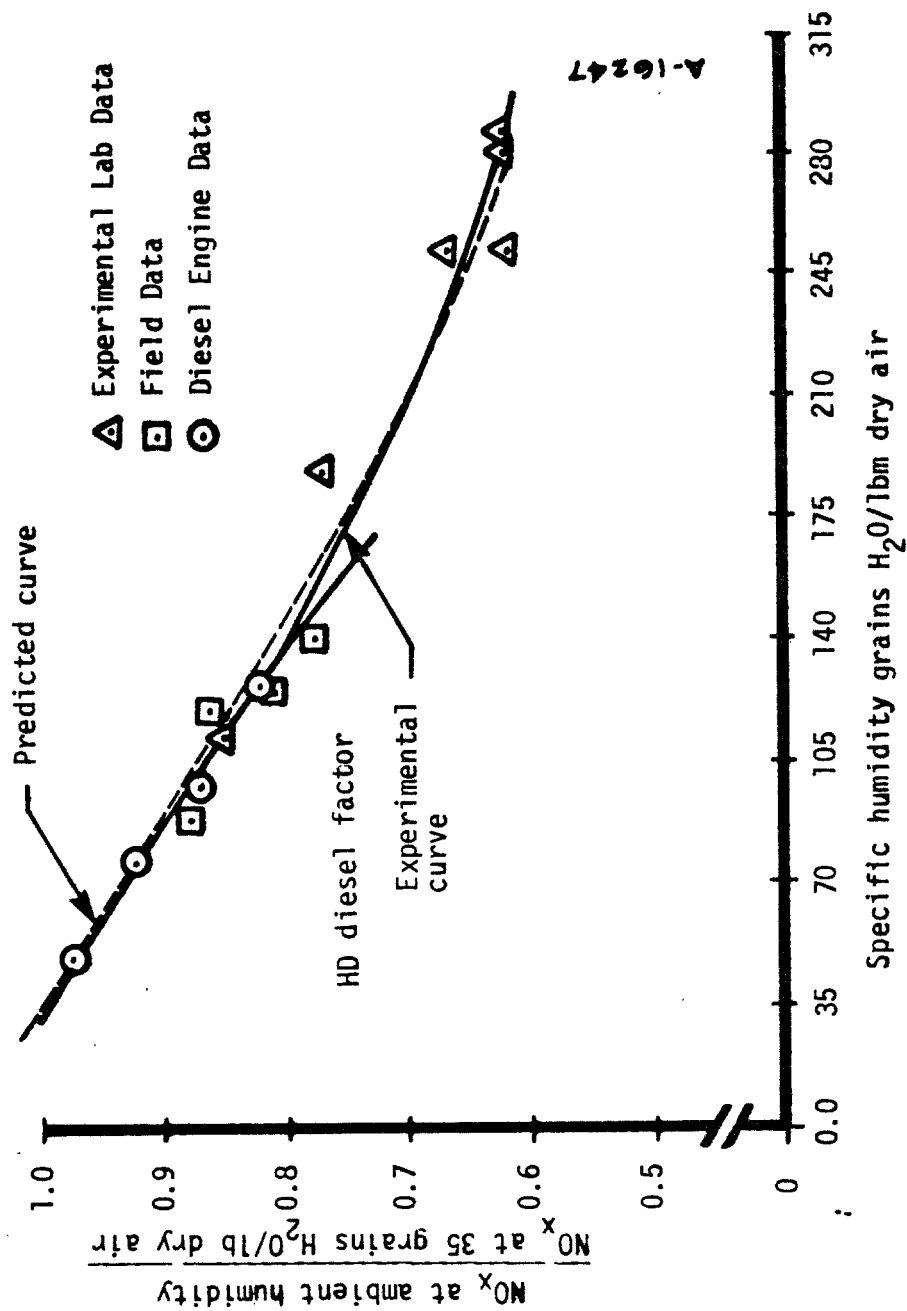


Figure C-14. Observed ambient humidity influence on NO_x production⁽⁶⁶⁾.

Gas turbine temperature corrections were also examined for application to reciprocating engine data. Figure C-15^(68,69) is a comparison of the HD diesel (4-TC, AC) correction factor with various gas turbine factors. Obviously, there is little agreement with the exception of one factor. Presumably these differences arise from differences in inlet air compressor and combustor design. The gas turbine factors, however, have some flexibility in that they are related to the combustor inlet parameters of pressure and temperature. Therefore, an attempt was made to relate changes in NO_x formation of reciprocating engines to changes in ambient temperature by estimating the cylinder temperature before combustion in a typical reciprocating engine and using the compressor pressure ratio and the assumption of isentropic compression to calculate temperature. This calculated temperature was then used with the gas turbine factors. Figure C-16^(70,71) presents the results of this approach for several different forms of gas turbine factors. Again, these results do not correlate well with the HD diesel factor, probably due to the empirical nature of the gas turbine equations and the large differences in air intake systems between engines and turbines.

On the basis of this brief review, gas turbine ambient correction factors do not make a useful contribution as potential ambient correction factors for reciprocating engines.

C.2.4 An Analytical Approach for the Ambient Temperature Correction of NO_x Emissions

Since no temperature correction factor has been reported in the literature (see Section C.2.1) for SI engines, and no systematic emissions data exist (either SI or CI) from which to base a temperature correction for large-bore engines in general, an attempt was made to develop a

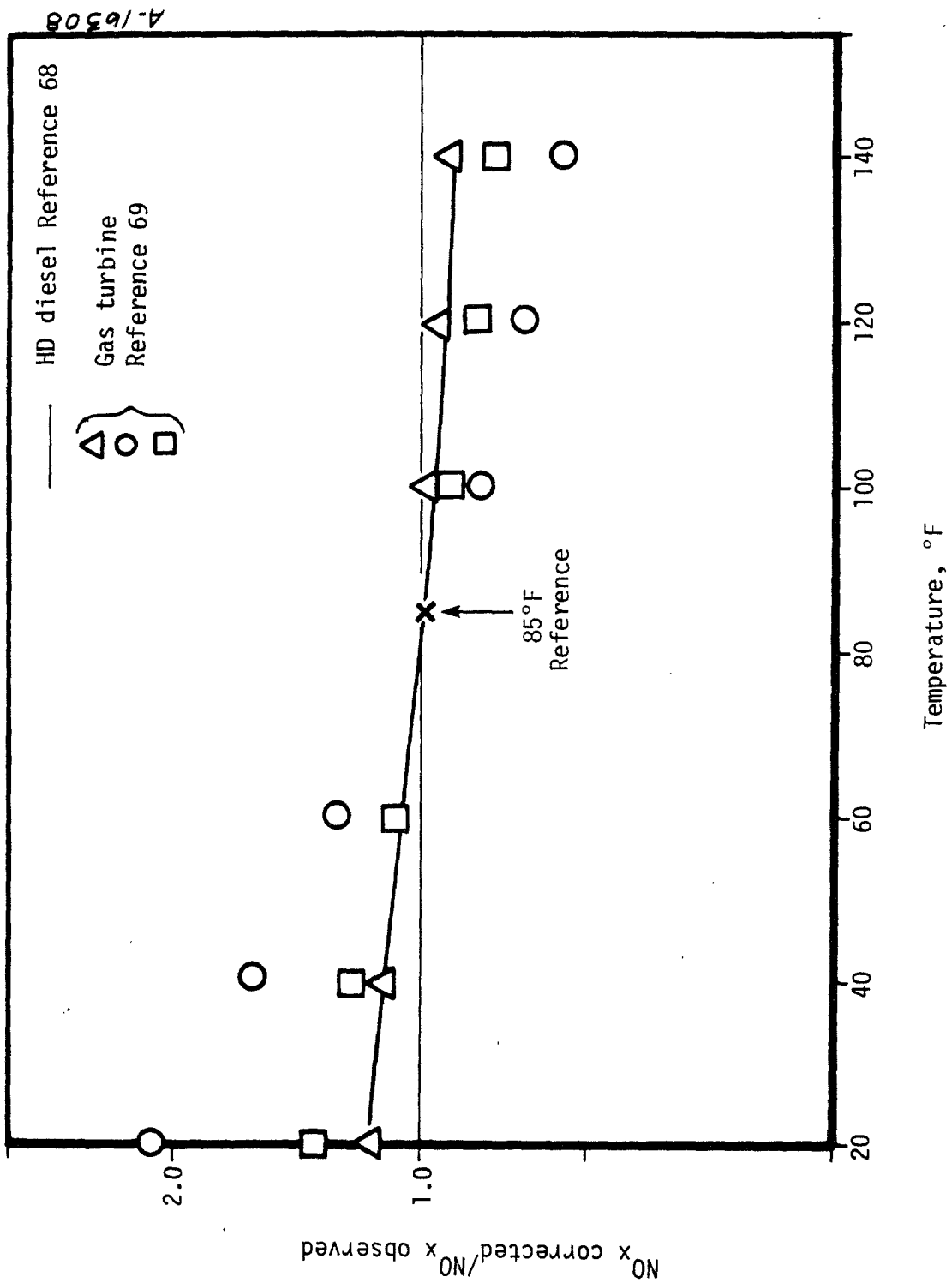


Figure C-15. Comparison of existing gas turbine ambient temperature correction with HD diesel temperature correction.

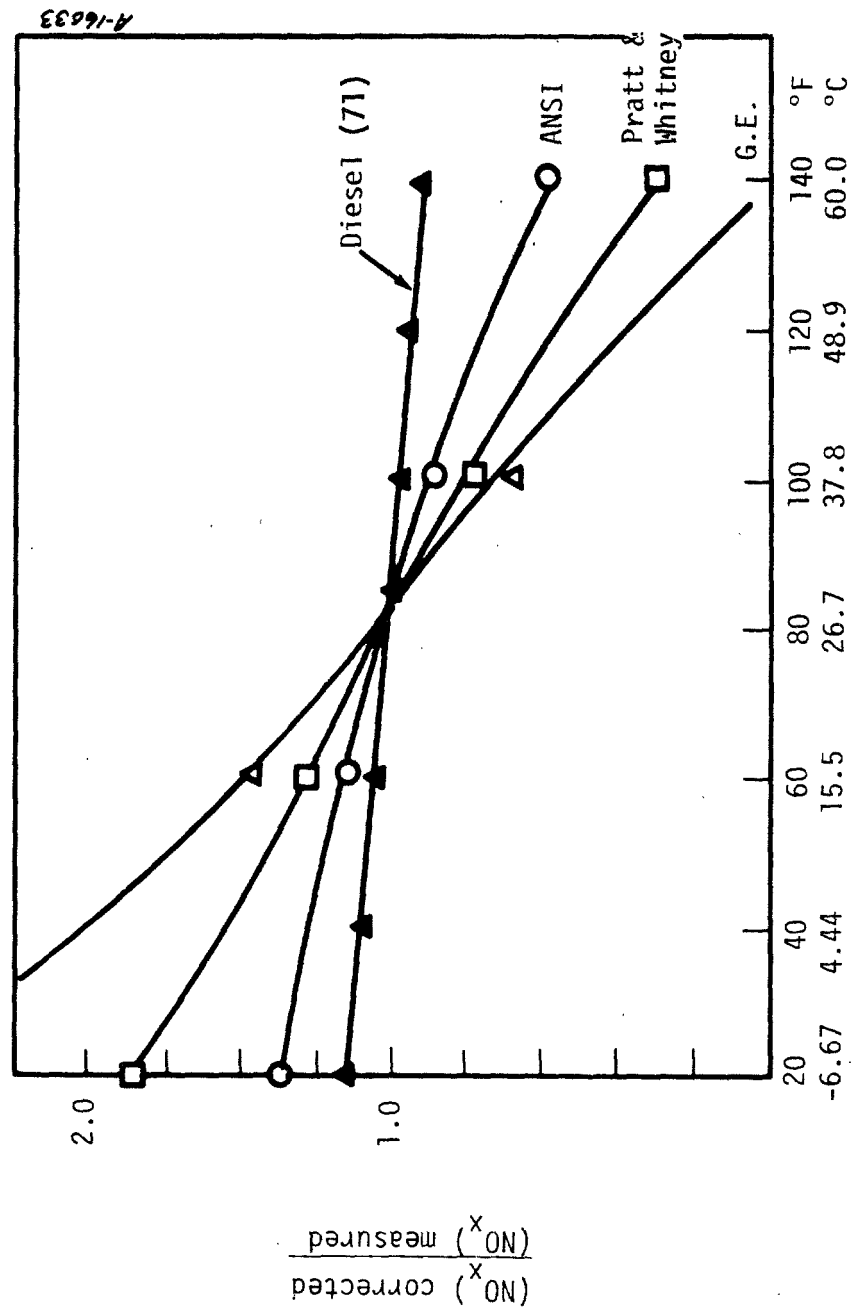


Figure C-16. Gas turbine temperature correction factors (70).

semianalytical approach for relating changes in ambient temperature to changes in NO_x level from large-bore engines. This approach was used to estimate how changes in ambient temperature affect NO_x levels.

The analytical approach is based on the fact that ambient temperature changes affect NO_x emissions by their direct effect on both fuel/air (F/A) ratio and peak flame temperature. As the ambient temperature rises, the inlet air becomes less dense. Since the air intake volume is essentially constant, the engine will inject a smaller mass of air; that is, the F/A ratio will increase. This change in F/A ratio is related to a change in NO_x level for both SI and CI engines.

The temperature of the air in a diesel engine or fuel/air mixture in a dual-fuel or natural-gas engine after compression is correspondingly changed by a change in ambient (inlet) air temperature. That is, an increase in inlet air temperature results in an even greater charge temperature after compression. This increase in charge temperature leads to a higher peak flame temperature and, consequently, greater NO_x levels. The increase in NO_x level due to an inlet air temperature increase is related to specific engine design parameters such as F/A ratio, degree of aftercooling, and compression ratio, as well as the fundamental nature of the combustion process (i.e., CI or SI). Therefore, the relationship between NO_x level and inlet air temperature is anticipated to be highly dependent on the particular engine design.

The discussion above indicates that NO_x level is primarily a function of both F/A ratio and inlet air temperature. That is,

$$\text{NO}_x = g(f, T)$$

where $f \equiv F/A$, fuel/air ratio

$T \equiv T_a$, ambient inlet temperature

$N \equiv NO_x$, oxides of nitrogen

The subscript R denotes a reference ambient condition

Since we are interested in predicting changes in NO_x level rather than absolute NO_x level we may mathematically represent a change in NO_x to a change in ambient temperature as

$$dN = (\partial N / \partial f)_T df + (\partial N / \partial T)_f dT$$

or

(C-4)

$$dN/dT = \partial N / \partial f df/dT + \partial N / \partial T$$

where the derivatives on the right side of Equation (C-4) remain to be evaluated. The first partial derivative, $\partial N / \partial f$, represents the change in NO_x emissions due to a change in F/A at constant ambient temperature, while the second partial, $\partial N / \partial T$, represents the change in NO_x due to temperature variations at constant F/A . This mathematical formulation can be portrayed graphically as shown in Figure C-17. The diagram shows NO_x vs. fuel-to-air ratio, the downsloping curves, and load vs. fuel-to-air ratio, the upsloping curves, for a typical turbocharged diesel engine. The basic problem is to find the change in NO_x at constant load due to some ambient temperature change. For example, assume the ambient temperature, T , is greater than the reference temperature, T_R . Starting at the uncorrected fuel-to-air ratio, one locates Point A on the NO_x production curve and Point B on the load curve. Then moving over, at constant load, to the reference temperature load curve, one locates Point C and, hence, the reference fuel-to-air ratio. Now,

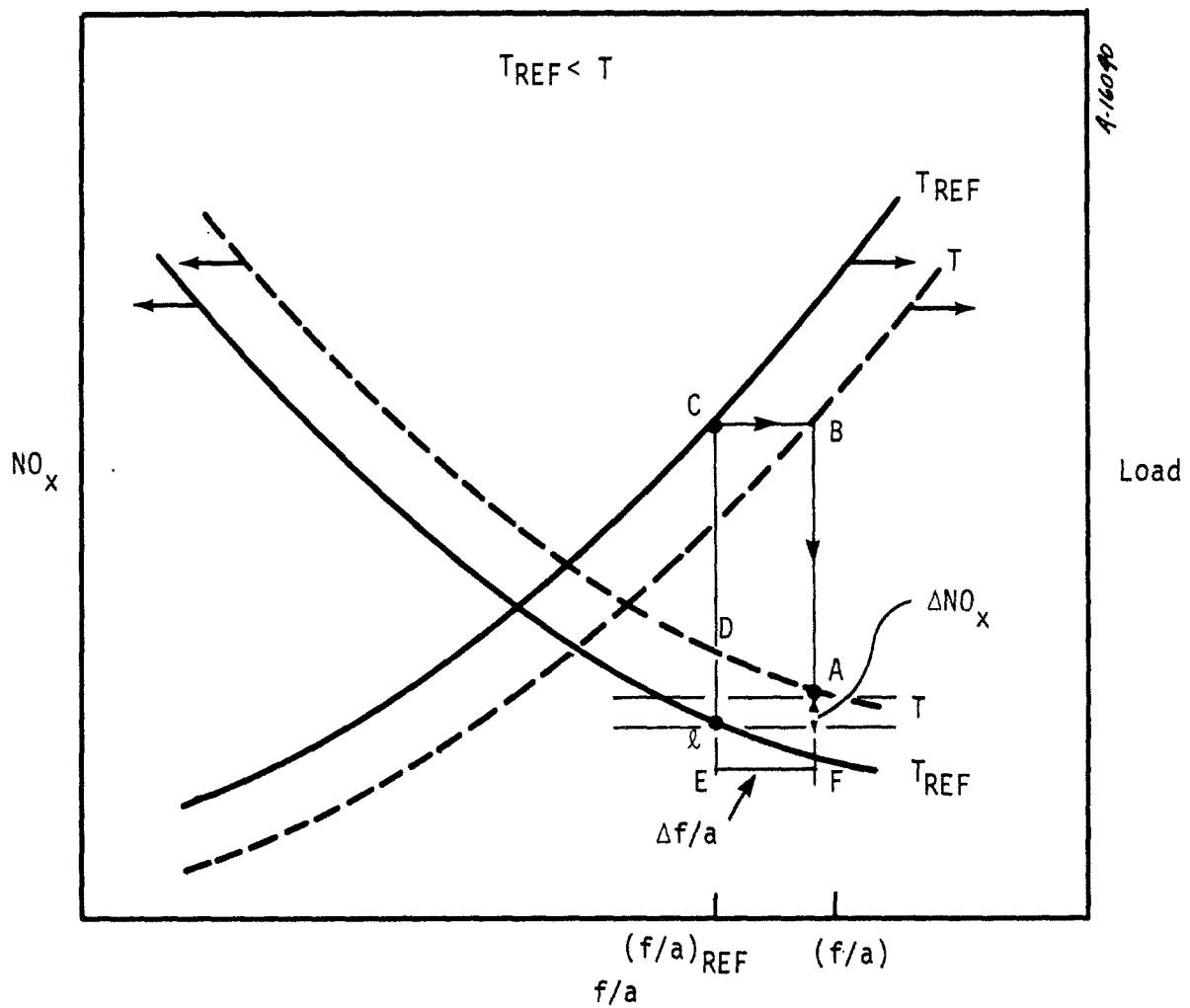


Figure C-17. Relationship of NO_x level and load to f/a ratio for a turbocharged diesel engine operating at two ambient temperatures.

knowing the reference fuel-to-air ratio, one can move from Point D to Point E on the reference temperature curve. Thus, the change in NO_x production resulting in this change of ambient temperature is indicated as ΔNO_x .

The change in NO_x level with F/A for a constant ambient temperature, $\partial N/\partial f$, is presumed known for a given engine or engine type. This relationship can be established from data that can be obtained in the laboratory on a sample engine. In addition, it is assumed that the NO_x vs. F/A plots of Figure C-12 have similar slopes for different ambient temperatures. Therefore, only NO_x vs. F/A data at one ambient temperature are required to evaluate the derivative.

The derivative df/dT , expresses the rate of change of F/A ratio with a change in ambient temperature. Wu and McAulay have derived the following expression relating F/A to ambient temperature (at constant load) for both turbocharged and naturally aspirated engines⁽⁷²⁾.

$$f = f_R (T/T_R)^B$$

and

(C-5)

$$B = (1 + n) - (1 - e_i) n (k - 1/\eta_K)$$

where

$k = C_p/C_v$, ratio of specific heats

η = turbocharger adiabatic compressor efficiency

n = turbocharger exponent from equation $T_A^n r_c = \text{constant}$ where
 $0.5 < n < 1$ depending on turbocharger compressor pressure ratio

r_c = turbocharger compressor pressure ratio

$e_i = (T_c - T_m)/(T_c - T_a)$

where T_c , T_m are compressor exit and intake manifold inlet temperature, respectively. Then, differentiating Equation (C-5) with respect to inlet temperature we can evaluate the term df/dT in Equation (C-4).

$$\begin{aligned} df/dT &= B(f_R/T_R) (T/T_R)^{B-1} = B(f_R/T_R) (1 + (\Delta T/T_R))^{B-1} \\ &\approx B(f_R/T_R) (1 + (B - 1) (\Delta T/T_R)) \end{aligned} \quad (C-6)$$

Since B is always less than 2 and T_R is $70^\circ\text{F} = 530^\circ\text{R}$, a nominal change in ambient temperature of 25°F makes

$$(B - 1) T/T_R \leq 1/20$$

Since this term is small compared to unity, we can approximate the derivative by

$$df/dt \approx B (f_R/T_R)$$

Thus, the term from Equation (C-4) that predicts the change in NO_x due to a change in F/A is evaluated as

$$\partial N/\partial f \, df/dt = (\partial N/\partial f) (B) (f_R/T_R) \quad (C-7)$$

An expression is now required to relate the change in NO_x level to a change in ambient inlet air temperature, i.e., the partial derivative $\partial N/\partial T$ of Equation (C-4). This dependence can be evaluated by first relating changes in ambient temperature to changes in flame temperature using the following relationship from Williams' Combustion Theory⁽⁷³⁾ suggested by Wilson, Muir, and Pellicciotti⁽⁷⁴⁾.

$$T_{f1} = T_B + 1/C_p ((Q - L) Y_{ox}/i + C_p(T_\infty - T_B))/(1 + Y_{ox}/i) \quad (C-8)$$

where $T_{f1} \equiv$ droplet diffusion flame temperature⁽⁷⁵⁾

$T_\infty \equiv$ isentropic compression temperature

$\equiv T_M(CR)^k$

$CR \equiv$ compression ratio of engine

$T_M \equiv$ manifold air temperature, related to T_A by degree of turbocharging and aftercooling

$T_B \equiv$ boiling point temperature of the fuel

$Q \equiv$ heat of combustion per unit mass of fuel

$L \equiv$ latent heat of vaporization of the fuel

$Y_{ox} \equiv$ ambient oxygen mass fraction

$i \equiv$ stoichiometric oxygen-to-fuel ratio (by mass)

$C_p \equiv$ specific heat of the fuel/air mixture

Then, the change in flame temperature is related to a change in NO_x production using the Arrhenius relation

$$dNO_x/dt \propto \exp(-K/RT_{f1}) ; K/R = 123,000^\circ R$$

If it is assumed that the rate of NO_x production is independent of time, one can readily integrate the Arrhenius equation to yield the following expression for the ratio of NO_x produced at a given ambient temperature, T , and a corresponding flame temperature, T_{f1} , to that at the reference condition.

$$N/N_R = \exp (K(T_{f1} - T_{f1R})/R(T_{f1} \times T_{f1R})) \quad (C-9)$$

The partial derivative can then be approximated in finite difference form by

$$\partial N/\partial T \approx N(1 - N_R/N)/(T - T_R) \quad (C-10)$$

Now by substituting Equations (C-7), (C-9), and (C-10) into Equation (C-4) we obtain:

$$\frac{dN}{dT} = \frac{Bf_R}{T_R} \frac{\partial N}{\partial T} + \frac{N \left\{ 1 - \exp \left[\frac{K}{R} \frac{(T_{f1} - T_{f1R})}{T_{f1} \times T_{f1R}} \right] \right\}}{T - T_R} \quad (C-11)$$

A preliminary evaluation of this expression for a turbocharged diesel engine predicts a 1-percent change in NO_x per $10^\circ F$ change in ambient inlet air temperature. For a constant intercooler effectiveness and turbocharger compressor efficiency, a 1-degree change in ambient temperature corresponds to a 1-degree change in manifold inlet temperature. Therefore, this analytical expression can be checked using NO_x emissions data vs. manifold air temperature (at constant load) for a turbocharged, diesel engine.

Emissions data for the manifold air cooling control (see Figure 4-34 of the draft SSEIS) indicates that this technique produces a 0.1- to 0.3-percent change in NO_x per degree Fahrenheit for diesel engines. Emissions data reported by Ingersoll-Rand (and from Figure 4-34 of the draft), show about a 1-percent change in NO_x per $10^\circ F$ change in manifold air temperature for turbocharged SI engines⁽⁷⁶⁾. Since NO_x emissions from SI engines are more responsive to changes in manifold air temperature, it appears that this analytical approach overestimates the effect of inlet air temperature on NO_x emissions from this turbocharged CI engine. The assumption of a constant NO_x production rate with peak flame temperature may be more valid for SI engines than CI engines owing to differences in the combustion processes.

The approach outlined here has the potential to incorporate many different parameters into a single correction factor. Additional terms could be added for any other operating parameters that have a significant effect on NO_x emissions, either directly or by their effect on F/A or T_{f1} . Once the appropriate relations are established, it may be possible to predict changes in NO_x levels as a result of changes in any one of these variables caused by ambient variations.

These goals are somewhat ambitious; therefore, the most logical first step is to gather available data taken at different ambient conditions and apply the method to prove its validity. At this time, however, there is insufficient information for either SI or CI engines to substantiate this approach.

C.3 REVIEW OF NO_x MEASUREMENT METHODS

Previous studies have indicated that sampling instrumentation and procedures significantly affect emission levels. Furthermore, no one standard procedure has been adopted by all of the manufacturers of stationary IC engines reporting emissions. That is, eight large-bore engine manufacturers who reported emission data used either chemiluminescent (CL) or nondispersive infrared (NDIR) and ultraviolet (NDUV) instruments and one of four emission measurement procedures (SAE, EPA, EMD, or DEMA).^{2/} Therefore, this section will present a detailed discussion of the instrument and sample acquisition practices used by the engine manufacturers. The purpose of this discussion is to identify possible variations in exhaust emission levels attributable to particular measurement equipment and/or procedures. Specifically, this section will:

- Establish which instruments, sampling trains, and procedures were used by each manufacturer who reported NO_x emissions
 - Summarize the potential sources of uncertainty relating to each measurement practice
 - Evaluate the variability among manufacturers' emissions data due to
- the following discussion will begin by illustrating typical measurement uncertainties in present sampling practices. This analysis will show that significant uncertainties in measurements can arise due to differences in both instrument and sampling procedures. Therefore, both of these sources of error will be discussed as they apply to the instruments and procedures used by the

^{2/}Alco, Cott, Cooper, DeLaval, Electro Motive, Ingersoll-Rand, Waukesha, and White Superior.

eight large-bore engine manufacturers. Then these practices will be compared and the uncertainty in each manufacturer's NO_x measurements estimated.

C.3.1 Previous Studies of Exhaust Measurement Variation

Before evaluating each manufacturer's sampling practice, it is of interest to examine the present "state-of-the-art" of internal combustion engine exhaust measurements of NO_x. Three recent studies have been conducted to compare the measurements of NO_x made by different laboratories from the same emission source. In two of these studies, NO_x measurements were made, simultaneously, in the same laboratory using identical procedures. In the other study, the same emission source was sent to each laboratory. These studies illustrate the magnitude of emission variations attributable to instruments and sampling procedures. They also indicate the reproducibility of emissions measurements, both within a laboratory and among different laboratories. The results of these studies will serve as a basis for comparing potential data variations due to the measurement practices of the eight large-bore engine manufacturers considered in the current study.

A series of cooperative emissions tests was conducted by the Coordinating Research Council (CRC) to evaluate measurement methods used to analyze diesel exhaust emissions from truck-size engines. In Phase III of this program, six laboratories sent sampling teams to one location to make simultaneous measurements (using NDIR analyzers) of a multicylinder engine⁽⁷⁷⁾. The engine used in this study was a six-cylinder, 300-cubic-inch, four-stroke, direct-injection diesel. The procedures that were used during this test program to measure NO, CO, and CO₂ evolved into SAE Recommended Practice J-177⁽⁷⁸⁾. In Phase IV of that program, the same engine was circulated to 15 laboratories to: (1) verify that the generally good

agreement of Phase III emission levels were the result of improved sampling procedures, and (2) obtain NO/NO_x data using CL analyzers(79).

Table C-84(80,81) shows the results of these two cooperative tests. Although the Phase III variations appear reasonable, the Phase IV results indicate poor agreement among the laboratories. These larger uncertainties were attributed to poor calibration procedures (span gases out of spec, instruments not calibrated) and possibly some variation in engine performance. In addition, it was noted that the average NO concentration measured at rated load by the CL analyzers was approximately 23 percent lower than that measured by NDIR.

A more recent cooperative test program conducted by the CRC evaluated EPA's revised heavy-duty diesel engine NO/NO_x measurement methods and instrumentation(82). Six participants made a series of NO/NO_x measurements on a multicylinder engine, simultaneously, and produced the range of uncertainty shown below. A range is shown because the results were analyzed in three different groupings: (1) all data, (2) those which remained after eliminating questionable results from participants who encounter sampling problems, and finally, (3) those which were left after excluding both questionable data and results obtained from instruments with long sample transfer times.

	13-mode	Standard Deviation	Standard Deviation
	(g/hp-hr)	(g/hp-hr)	(Mean, %)
NO	8.03 - 8.21	0.42 - 0.29	5.9 - 3.5
NO _x	8.05 - 8.16	0.30 - 0.17	3.8 - 2.1

All but one of the participants used a CL analyzer, and the results from the one NDIR analyzer (a new reduced interference design with a NO_x converter)

TABLE C-84. ACCURACY OF NO_x MEASUREMENTS (80,81)

Source	Phase III 6 Labs, Simultaneous Measurements		Phase IV 15 Labs, Round Robin	
	Standard Deviation as % of Average	Spread in Data as % of Average	Standard Deviation as % of Average	Spread in Data as % of Average
NO: NDIR CL	5	10	8	27
	—	—	36	15
NO ₂ : NDUV CL	—	—	21	39
	—	—	88	166

were equivalent to those from the CL analyzers. Thus, based on simultaneous tests using the same instruments, sampling practices, and emissions sources, uncertainties in emission measurements among laboratories ranged from 3 to 6 percent for NO, and 2 to 4 percent for NO_x. In addition, the repeatability within a laboratory ranged from 2 to 7 percent for NO and 1 to 3 percent for NO_x.

In contrast to the above program, the eight large-bore engine manufacturers used one of four test procedures (SAE, EPA, DEMA, or EMD) and either CL or NDIR/NDUV instruments. Furthermore, the NDIR's were not reduced interference designs. Hence, their data are likely to vary more due to measurement practices than the truck engine results. The following discussion will first examine the sources of such variations from differences in both instruments and sampling practices. Then an attempt will be made to suggest uncertainty bounds for the emissions data from each manufacturer.

C.3.2 Variations in NO_x Emissions Related to the NO_x Instrument

Since the development of commercial chemiluminescent analyzers, (1971), various studies have been conducted to compare their operation with the already established NDIR analyzer. All of these studies have shown the NDIR analyzer to record consistently higher levels of NO than the CL analyzer for a given source(83-85). Three of the large-bore engine manufacturers reported NO_x emissions using NDIR's and the other five used CL analyzers.

Table C-85 shows the NO_x instruments used by each of the manufacturers. Scott Research Laboratories made the emission measurements for Ingersoll-Rand and White-Alco, since neither manufacturer presently owns emission measurement equipment. Note that an electrochemical instrument was used by Shell Oil Research in 1971 to measure NO_x from one Cooper-Bessemer

TABLE C-85. INSTRUMENTS USED BY MANUFACTURERS TO MEASURE NO_x EMISSIONS

Manufacturer	CL (with Thermal Converter) NO/NO _x	NDIR/NDUV NO, NO ₂ /NO ₂	Electrochemical NO/NO ₂
Colt	Thermo Electron 10A		
Cooper Energy	Thermo Electron 10A	NO: Beckman IR 18A	Envirometrics NS-200
DeLaval	Scott 125		
Ingersoll-Rand ^a	Scott 125		
Waukesha		NO: Horiba AIA-2	
White Superior (Div. Cooper Energy)	Scott 125		NO: Dynasciences NX-130 NO ₂ : Envirometrics NS-200
White-Alco ^a	Scott 125		
GMC/EMD		NO/Beckman 315B NO ₂ /Beckman 255	

^a Measurements made by Scott Environmental Technology Laboratory.

and one Ingersoll-Rand engine. None of the manufacturers has used this instrument since then; therefore, no attempt will be made to correlate those emissions with instrument differences. Only Waukesha and GMC/EMD continue to measure NO_x emissions with NDIR analyzers, and Waukesha has recently acquired a CL analyzer. Thus, all but one manufacturer will be using CL instruments in the future. In the following paragraphs, the principle of operation of the NDIR and CL instruments and associated sources of error will be briefly reviewed. Then the results of NDIR/CL comparisons will be summarized to seek a method of expressing emissions on an "equivalent instrument" basis.

C.3.2.1 NDIR Instrumentation

The NDIR instrument was introduced nearly 20 years ago and has continued to be used widely as a CO and CO_2 detector. In addition, it was used extensively to detect NO, HC, and SO_2 , but other methods are not supplanting it for these species. Its principle of operation depends upon absorption of infrared radiation by the gaseous sample. Figure C-18 illustrates a typical NDIR instrument. Built-in optical and gaseous filters are used to produce a narrow infrared beam band width to compensate for interference (absorption) by other constituents.

Despite these precautions, water vapor and CO_2 , to some extent, may cause positive interferences (high readings) even though refrigerant and chemical driers are used to remove water vapor. Desiccants, however, have been found to cause significant interferences as well as water vapor (see later discussion and Section C.3.3). In addition to these problems, the response of the NDIR instrument to the specie of interest is nonlinear in some instrument designs, necessitating a carefully constructed calibration curve using at least four, and preferably six to eight, calibration gases.

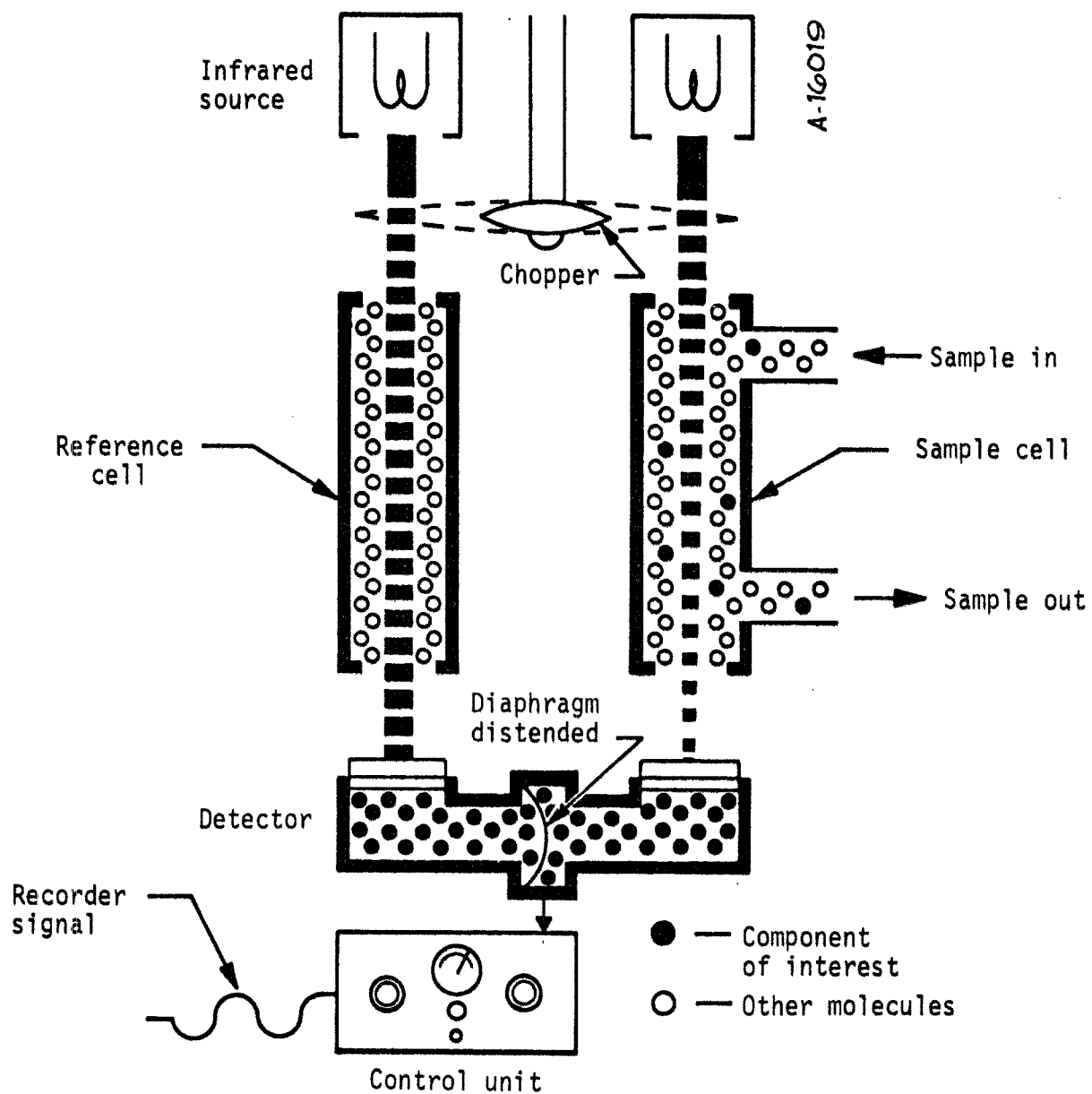


Figure C-18. Nondispersive infrared absorption analyzer.

C.3.2.2 Chemiluminescent Analyzers

Chemiluminescent analyzers, in contrast to NDIR instruments, have been developed only recently (1971) for source sampling. Nevertheless, the CL instrument has gained increasing application for the measurement of NO_x ($\text{NO} + \text{NO}_2$) and NO . In this type of instrument ozone, O_3 , is reacted with nitric oxide, NO , to produce a chemically excited state of NO_2^* , which emits light as it decays to stable NO_2 . The intensity of this emitted light is proportional to the NO concentration present in the sample. The analyzer uses a photomultiplier to detect the light.

Figure C-19 is a schematic of a CL analyzer and illustrates the reaction of ozone with NO in the reaction chamber of the instrument. The instrument is easier to calibrate than some NDIR's due to its linear response to NO , and thus requires fewer calibration gases. Note that a $\text{NO}_2 \rightarrow \text{NO}$ converter is depicted in Figure C-19. There are basically two types on the market. One device consists of a stainless steel tube which is heated to $\approx 1200^\circ\text{F}$ and essentially converts all NO_2 in the sample to NO . The other device catalytically converts NO_2 to NO . In this way just NO or both NO and NO_2 can be measured, depending on whether the sample gas is fed directly to the reaction chamber or passed through the converter first. The NO_2 level is then deduced by subtracting the NO level from the total NO_x reading.

Potential problems encountered in this instrument include quenching of the excited NO_2 by other species, converter inefficiencies, and interferences caused by chemiluminescence of other gases. In general, quenching (by CO_2 or H_2O) is negligible in CL instruments; particularly if water is removed from the sample before analysis and low pressure (vacuum) reaction chambers are used to reduce quenching by CO_2 . Nevertheless, some quenching problems have been observed during measurement of fuel-rich automotive exhausts. Quenching

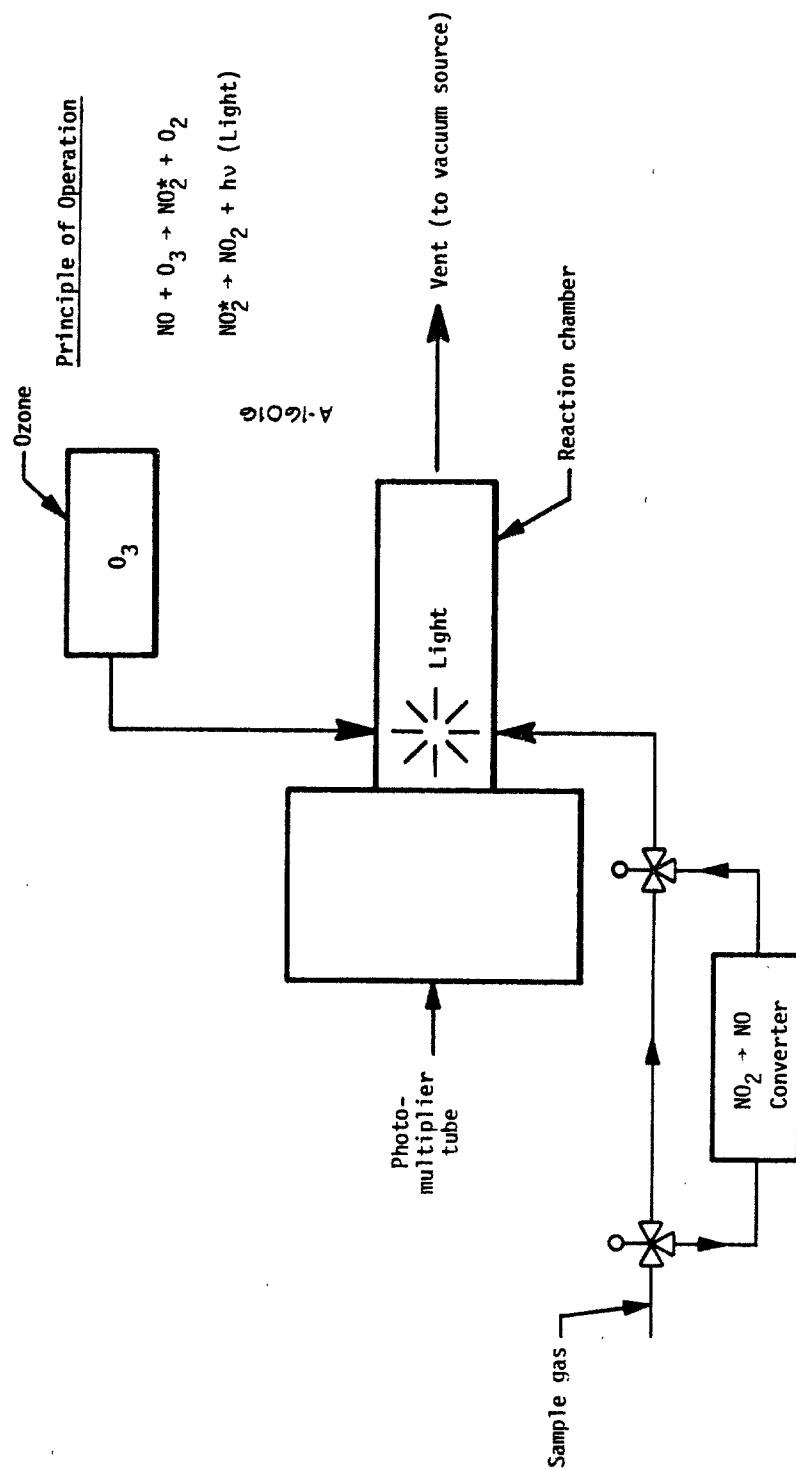


Figure C-19. Chemiluminescent analyzer.

effects, however, have not been observed during measurements on truck-size diesels, whose exhausts are similar to those of large bore stationary engines (i.e., ≈ 15 percent oxygen).

Converter problems are avoided by thermal conditioning of new converters and by making regular checks of converter efficiency (≥ 90 percent). Interferences are minimized by the instrument manufacturer through the choice of spectral filters and photodetectors.

C.3.2.3 Sources of Instrument Error

Sources of error for each of the instruments listed in Table C-85 are summarized in Table C-86⁽⁸⁶⁻⁹⁴⁾. Note that neither the quenching effect nor the converter problems of the CL analyzer should occur when measuring large-bore engine exhausts with a properly operated and maintained CL analyzer. Present NDIR analyzers, on the other hand, are susceptible to errors due to interference despite sample conditioning to remove water. This interference, combined with the relative ease of operation and accuracy of CL analyzers, has led to an increasing preference of CL's for NO_x measurements. As an example, EPA's proposed revisions to the Heavy Duty Diesel and Gasoline Engine Sampling Procedure specify CL analyzers. The following section summarizes comparisons of CL and NDIR/NDUV measurements of NO_x . These comparisons will be the basis for expressing NDIR/NDUV measurements as equivalent CL levels.

C.3.2.4 Correlations of CL to NDIR/NDUV

As discussed above, NDIR analyzers are subject to water vapor interference despite sample conditioning to remove water vapor. These interferences cause the instrument to indicate higher NO_x levels than are actually present. A comprehensive study conducted by TRW (with Scott

TABLE C-86. SOURCES OF NO_x INSTRUMENT ERROR

Chemiluminescent, NO/NO ₂	NDIR, NO (86,87)	NDUV, NO ₂ (88)	Electrochemical NO/NO ₂ (89)
<ul style="list-style-type: none"> • Quenching of CL by third body reactions^a (CO₂, H₂O) • NO₂/NO converter^b <ul style="list-style-type: none"> — NO_x → N₂ at high temperature, low O₂, high CO and HC — NO₂ → NO incomplete due to improper converter conditioning — NH₃ → NO, NO₂ at high temperature especially with some catalytic converters in rich exhausts — NO₂ → NO at low temperatures, low O₂ 	<ul style="list-style-type: none"> • Water vapor interference causes higher NO_x readings. Also CO₂ interference to some extent; optical filters and water removal minimize interferences • Use of chemical driers lead to NO₂ adsorption and NO₂ → NO conversion • Calibration curve non-linear, requires frequent checks using 4 to 6 calibration gases 	<ul style="list-style-type: none"> • Interference caused by carbon particles and other species, results in a drifting, high reading. 	<ul style="list-style-type: none"> • Not much application in source sampling • SO₂ interferences • Nonlinearity in 30 to 300 ppm range • Susceptible to error due to temperature changes

^aNot present in properly designed instruments (90-92).

^bCondition CL converter to avoid error; perform regular converter efficiency. Most of these errors occur in rich exhausts unlike the lean exhausts (~15 percent oxygen) of large bore engines (93-94).

Research Labs) using constant volume sampling (CVS) of automotive exhausts determined that

$$(\text{NO})_{\text{NDIR}} = 1.07 (\text{NO})_{\text{CL}} - 4.2 \quad (\text{C-12})$$

where (NO) denotes NO concentration in ppm.⁽⁹⁵⁾ Both analyzers had similar water removal systems (condenser and chemical driers); hence, the difference was due to a gas interference only. In addition to this finding, the study established that chemical driers promoted $\text{NO}_2 \rightarrow \text{NO}$ conversion. Comparison of an NDIR with water removal to a CL without water removal indicated that

$$(\text{NO})_{\text{NDIR}} = 1.12 (\text{NO})_{\text{CL}} + 8.4 \quad (\text{C-13})$$

Furthermore, a comparison of two CL analyzers, one (CL_B) preceded by a water trap and chemical drier and the other (CL_A) without either, indicated that

$$(\text{NO}_x)_{\text{CL}_B} = 0.873 (\text{NO}_x)_{\text{CL}_A} + 6.6 \quad (\text{C-14})$$

Thus, the net effect of the drier and water trap is to destroy about 12 percent of the NO_x in the sample.

The results of this study and others that corroborate these effects are summarized in Table C-87⁽⁹⁶⁻¹⁰⁰⁾. The bias of NDIR analyzer ranges from 3 to 30 percent higher based on these studies. As this table indicates, the particular correction that applies to any one of the three manufacturers that used an NDIR analyzer (and NDUV for NO_2) will depend on their sampling setup and procedures (e.g., type and location of water trap, and the NO_2 present). Therefore, the appropriate correction for NDIR instrument bias will be identified after a discussion of each manufacturer's sampling practices.

TABLE C-87. COMPARISON OF CL AND NDIR, NDIR/NDUV MEASUREMENTS

Source	Comparison	Comments
TRW Study (96) (Reference 5)	$\text{NO}_{\text{xCL}}/\text{NO}_{\text{xNDIR/NDUV}} = 0.82$	No water removal from CL sample.
	$\text{NO}_{\text{xCL}}/\text{NO}_{\text{xNDIR/NDUV}} = 0.75$	Water trap and drier used for both CL and NDIR.
Clemmens Memo (97)	$\text{NO}_{\text{xCL}}/\text{NO}_{\text{xNDIR}} = 0.9$ $(\text{NO}_{\text{CL}}/\text{NO}_{\text{NDIR}} = 0.88)$	Used TRW comparison for NO (no water removal for CL) and assumed that NO ₂ was 2 percent (on the average) of total NO _x in a diesel exhaust.
SAE 750204 Phase IV (Reference 98)	$\text{NO}_{\text{CL}}/\text{NO}_{\text{NDIR}} = 0.77$	Result questionable due to other sampling errors.
SAE 730213 (Reference 99)	$\text{NO}_{\text{xCL}}/\text{NO}_{\text{xNDIR/NUV}} = 0.94$	None of eight large-bore manufacturers used DUV ^a instrument.
CRC Report #487 (Reference 100)	$\text{NO}_{\text{xCL}}/\text{NO}_{\text{xNDIR}} = 0.985$	NDIR instrument is "third" generation, low interference analyzer, not comparable to NDIR's used by large-bore manufacturers.

^aDispersive Ultraviolet

C.3.3 Variations in NO_x Levels Related to Sampling Procedures

In addition to the choice of instrumentation, differences in sampling procedures can cause variations in reported emission levels. This was clearly illustrated in Table C-84 for the Phase IV cooperative tests where 15 laboratories made measurements of an identical source. Since four basic sampling procedures (DEMA, SAE, EMD, EPA) were used by the eight large-bore engine manufacturers, some variation between their reported emission levels can probably be attributed to differences in their sampling practices. The sources of these uncertainties will be identified and the four sampling procedures compared and evaluated in this section.

C.3.3.1 Sources of Sampling Error

The two major problems encountered in measuring engine exhausts are: (1) chemical changes that occur during transfer of the sample to the analyzer, and (2) error due to improper operation of the analyzer. Figure C-20 is a simple schematic of an engine sampling system. In transferring the exhaust gas sample to the analyzer, care must be taken to ensure that all of the NO_x (NO + NO₂) or NO (when only an NDIR is used) originating from the engine exhaust reaches the analyzer. For these reasons sample lines are heated to prevent condensation or kinetic conversion of constituents. High sample flowrates (short sample residence times) are maintained to minimize sample degradation during its transfer from the engine to the analyzer, and water removal devices are employed to minimize instrument interferences from water vapor. Similarly, it is important that the analyzer be calibrated and all components, such as NO₂ → NO converters (on CL instruments), be functioning properly. Therefore, adequate analyzer specifications and calibration procedures are essential for accurate emission measurements.

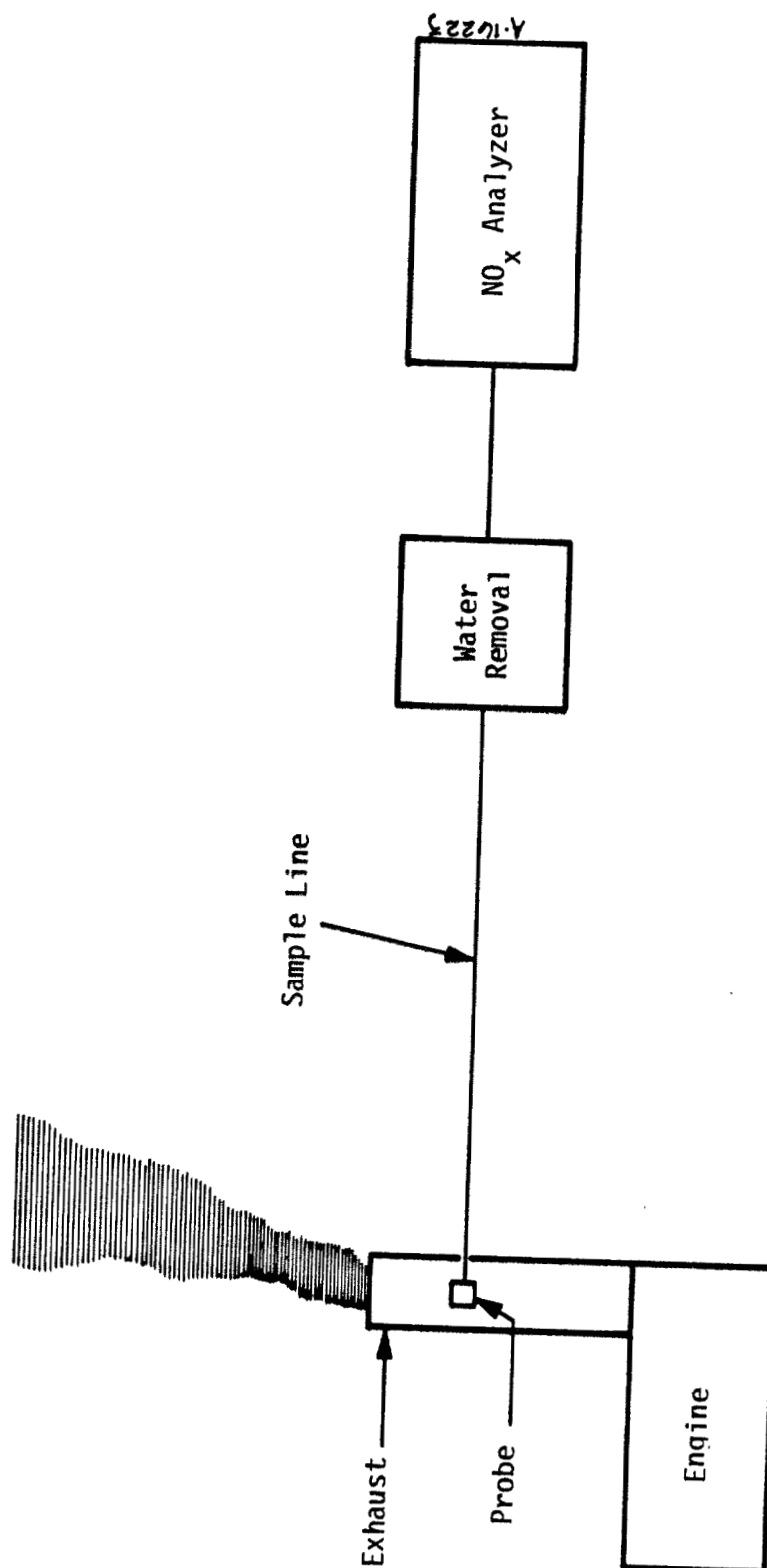


Figure C-20. Schematic of exhaust sampling system.

Table C-88(a)(101-107) presents a more detailed summary of sources of sample transfer error. As this table indicates, heated sampling lines coupled with low sample residence times are essential to preserve the initial amounts of NO and NO₂ contained in the sample gas. Furthermore, leak checks of the entire sampling system (particularly on the vacuum side) should always be performed to assure that the sample gas reaches the analyzer undiluted. Water removal is required with NDIR analyzers; therefore, care must be taken to minimize the time available for NO to be converted to NO₂ which can then be absorbed by the condensed water from the sample gas. Chemical driers (desiccants) are an unacceptable water removal device since they promote the NO₂ → NO reaction and absorb NO₂ (see Section C.3.2).

Table C-88(b)(108) summarizes important analyzer related procedures required for accurate measurements. Care must also be exercised in operating emission measurement instruments. Frequent calibrations should be performed using accurate, certified blends of calibration gases. Zero and span checks also serve to indicate potential instrument problems as well as necessary gain adjustments. Span gases should be frequently cross-checked with calibration gases or checked against a NBS standard since even certified gases can be in error.

Some calibration curves for NDIR analyzers are nonlinear; therefore, several calibration gases (minimum of six) should be used to calibrate the instrument(109). These instruments also experience some hysteresis after sitting unused; therefore, frequent calibration is necessary. Also, calibration points should be curve-fit using a higher order polynomial. In addition, efficiencies of NO_x converters (on CL analyzers) should be checked regularly. Finally, strip chart recordings of data are superior to visual

TABLE C-88(a). SOURCES OF SAMPLING ERROR: THE SAMPLE TRANSFER

Sampling System		
Source	Error	Correction
<ul style="list-style-type: none"> ● Unheated sampling line 	<p>NO → NO₂; NO₂ adsorbed during water removal: Observed 40-percent loss of NO when sample residence times were 35-40 sec (101)</p>	<p>Heat line to 375°F</p>
<ul style="list-style-type: none"> ● Long sample residence times 	<p><u>Heated Line:</u> NO₂ adsorbed during water removal: 6-7% NO_x loss for system response times^a greater than 25 sec (102)</p> <p><u>Cold Line:</u> 12-percent NO_x loss for sample residence time of 13 sec (103)</p>	<p>Use short sample line and/or flowrates to limit system response to 15 sec or less.</p>
<ul style="list-style-type: none"> ● Sample line connections and fittings 	<p>Leaks that dilute sample gas: Observed 25-percent loss of NO_x due to pre-filter leak (104)</p>	<p>Leak check system before testing</p>
<ul style="list-style-type: none"> ● Water removal <ul style="list-style-type: none"> — Refrigerant 	<p>NO₂ adsorbed in condenser; (105)</p>	<p>CL: Locate condenser after converter. NDIR: Maintain high sample flow through condenser, remove H₂O as it forms.</p>
<ul style="list-style-type: none"> — Chemical drier (desiccant) 	<p>NO₂ → NO and drier "eats" NO₂. Negative or positive errors with cold lines (SAE procedure) depending on how drier conditioned (106, 107)</p>	<p>Do not use chemical driers</p>

^aSystem response time = sample residence time + instrument response time

TABLE C-88(b). SOURCES OF SAMPLING ERROR: INSTRUMENT RELATED

Instrumentation		
Source	Error	Correction
<ul style="list-style-type: none"> Instrument drift, ozone shortage (CL), plugged capillary 	Low NO _x readings	Calibrate instrument frequently. Zero and span before and after each measurement for all instrument ranges. Instrument should meet minimum performance criteria.
<ul style="list-style-type: none"> Converter malfunction (CL) 	Not all NO ₂ converted to NO, results in low NO _x reading also possible for NO to exceed NO _x levels	Perform converter checks regularly. Use known standard to check converter efficiency.
<ul style="list-style-type: none"> Visual rather than stripchart readings 	Analyzer meter only accurate to ±3 percent. Visual averaging less consistent than chart averaging (108)	Use stripchart, average levels over an interval for which steady state conditions exist.
<ul style="list-style-type: none"> Calibration and span gases 	Change of constituents with time, or erroneous certification	Use certified gases of specified blends. Cross-check span and calibration gases.

analyzer meter readings since a permanent record is produced, and emission levels can be averaged over a time interval more accurately and consistently.

C.3.3.2 Sampling Procedures: EPA, DEMA, SAE, EMD

Having completed this brief review of sources of sampling error, we can examine each of the four measurement practices presently used by the eight large-bore engine manufacturers. Table C-89 is a comparison of the EPA, DEMA, SAE, and EMD emission measurement practices⁽¹¹⁰⁻¹¹³⁾. The information for the EPA procedures is based on the revised EPA Heavy Duty Diesel Emissions Regulations as proposed in the Federal Register, Volume 41, No. 101, May 24, 1976. This revised procedure requires CL analyzers for NO/NO_x measurement, establishes instrument and calibration specifications, and defines sample transfer configurations and practices.

This comprehensive EPA regulation was recently verified through cooperative testing by both manufacturers of mobile diesel engines and the EPA-Ann Arbor Mobile Sources Laboratory⁽¹¹⁴⁾. The six participants in this program (see Section C.3.1) made essentially equivalent NO_x measurements for a series of eight tests. Two participants, however, did experience a small, but consistent, NO/NO_x crossover (NO levels greater than NO_x levels). Nevertheless, the standard deviation of measured NO_x levels was generally small, ranging from 2 to 4 percent (of the mean level). Therefore, the EPA practice will be the basis of comparison for the other measurement practices.

The potential sources of measurement uncertainty of the other three procedures (DEMA, SAE, and EMD) are summarized in Table C-90. The sampling trains for these three procedures, as well as the EPA setup, are illustrated in Figure C-21. The DEMA procedure specifies a CL analyzer and the SAE/EMD

TABLE C-89. COMPARISON OF SAMPLING PRACTICES

- Test Procedure

DEMA: No time between modes specified
 EPA: 5 minutes
 SAE: 20 minutes
 EMD: 30 minutes

DEMA: Data recorded and averaged over 10 minutes
 EPA: Data recorded over last 2 minutes
 SAE: Data recorded over last 5 minutes
 EMD: Data recorded over last 3 minutes

- Instrument Calibration — DEMA and EPA are CL; SAE is NDIR; EMD is NDIR/NDUV

DEMA: Analyzers calibrated semimonthly. No details of calibration given.
 EPA: Check NO_x converter once per week (must be at least 90-percent efficient); leak check system, calibrate analyzer, check sample line residence time, and quench check every 30 days
 SAE: Calibrate monthly
 EMD: Checked and calibrated monthly, semiannually and annually

DEMA: Number of calibration gases not specified
 EPA: Span analyzer with calibration gases having nominal concentrations of 30, 60, and 90 percent of full scale concentrations
 SAE: Use calibration gases that are 25, 50, 75, and 100 percent of instrument range used.
 EMD: Use 4 calibration gases

DEMA: No specifications for calibration or span gas blends and dilutents
 EPA: Blends and dilutents specified and accurate within 2 percent of true concentration or traceable within 1 percent of NBS blends. Span gas traceable to within 1 percent of calibration gas.
 SAE: ±2 percent accuracy on gas analysis certification
 EMD: ±2 percent accuracy blends and dilutents defined by EMD

DEMA: No analyzer specifications required
 EPA: Specifications for response-time, precision, noise, zero and span drift, and linearity
 SAE: Accuracy should be ±2 percent full-scale deflection or better
 EMD: Same as SAE

TABLE C-89. Concluded

● Sample Transfer

DEMA: Stainless steel (S.S.) probe, configuration not specified
 EPA: Specifies number and size of holes in S.S. probe
 SAE: S.S. probe, no configuration specified
 EMD: S.S. probe, configuration specified

DEMA: Sampling line material not specified
 EPA: Stainless steel, teflon, or proven inert
 SAE: Stainless steel or teflon after exhaust cooled
 EMD: Stainless steel

DEMA: Line heated to 200°F for NO_x (375°F for HCT)
 EPA: 375°F + 18°F - 9°F
 SAE: Unheated
 EMD: Unheated

DEMA: No sample line length specified
 EPA: <50 feet or sample line residence and instrument response time is less than or equal to 15 seconds
 SAE: No length specified
 EMD: No length specified

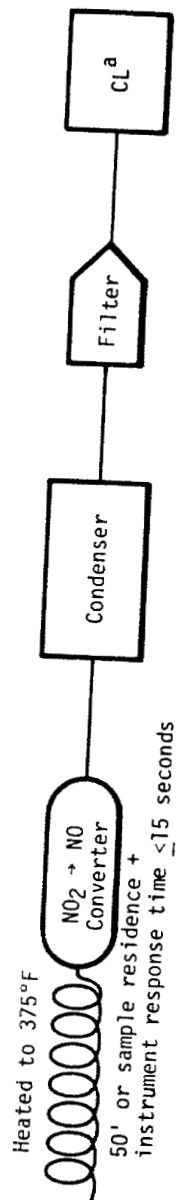
DEMA: Chemical driers can be used
 EPA: Not allowed
 SAE: Refrigerant and chemical drier specified
 EMD: Same as SAE

DEMA: Recommend water removal at probe or at analyzer before NO₂ → NO converter
 EPA: Condenser at analyzer after converter
 SAE: Condenser and drier before analyzer
 EMD: Same as SAE for NDIR. NDUV does not use water removal.

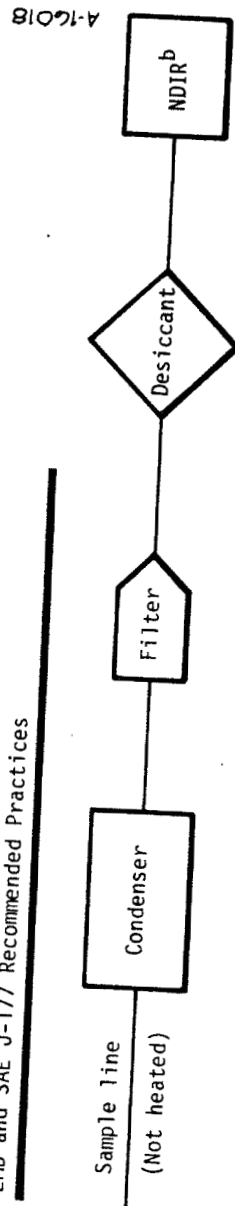
TABLE C-90. SOURCES OF UNCERTAINTY FOR DEMA, SAE, EMD EMISSION PRACTICES

DEMA (CL)	EMD (NDIR/NDUV)/SAE (NDIR)
<ul style="list-style-type: none"> • Unheated sampling lines permitted • No specification of sample residence time in sampling line, or system response time • Leak checks are not specified • Water removal device can be located at analyzer, but before $\text{NO}_2 \rightarrow \text{NO}$ absorption in water trap • Chemical driers permitted • No instrument specifications • Converter efficiency checks not specified • No calibration procedures specified • Calibration and span gas specifications not defined 	<ul style="list-style-type: none"> • Unheated sample lines • No sample residence or system response time specified • Leak checks not specified • Allow chemical drier • Calibration procedures not specific (e.g., what constitutes out-of-calibration, how calibration points are curve fit, etc.) • Calibration and span gas blends and diluents not specified by SAE. EMD has "own" specifications.

EPA-HD Diesel and Gasoline (Proposed) Regulations

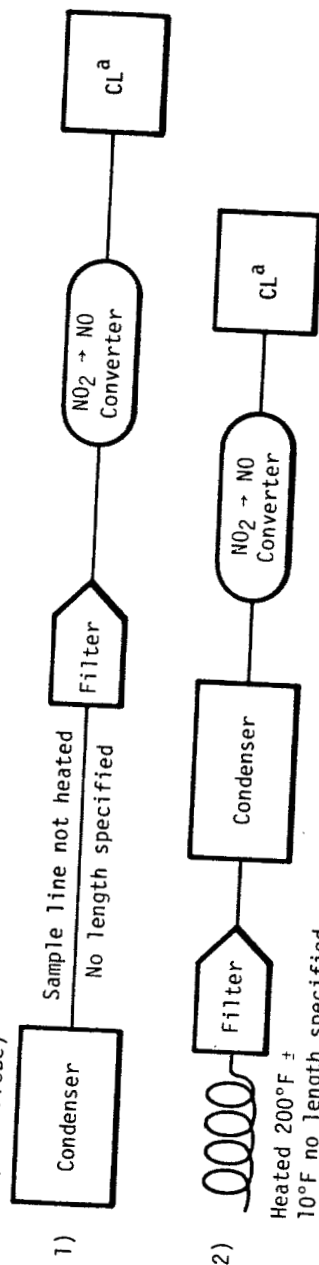


EMD and SAE J-177 Recommended Practices



DEMA SETUP

(Near Probe)



^aCL - Chemiluminescent analyzer

^bNDIR - Nondispersive infrared analyzer

Figure C-21. Sampling practices.

procedures require an NDIR instrument. The EMD practice also utilizes an NDUV analyzer, in series after the NDIR, to measure NO_2 in the sample.

Table C-90 clearly indicates that there are several sources of measurement uncertainty possible for the DEMA procedure, largely as a result of its failure to define instrument and sample transfer practices. Its most obvious shortcoming is the potential degradation of the NO and NO_2 in the sample gas as it passes from the probe through an unheated sample line (of unspecified length) and condenser before entering the CL analyzer. High residence times would also promote NO_x loss for this configuration. One large-bore engine manufacturer demonstrated that the sample residence time must be less than 3 seconds to prevent significant loss of NO_2 in the sample (12-percent loss for a 13-second residence time) as it passed through a cold sample line and condenser (see Table C-88).

Similarly, SAE and EMD practices can also lead to a loss of NO_x as the sample passes through an unheated sample line, a condenser, and a desiccant before reaching the analyzer. As discussed previously, the desiccant converts NO_2 to NO and absorbs NO_2 .

Comparisons of the SAE sampling train with the proposed EPA train indicated that SAE measured concentrations of NO are understated by 15 to 40 percent during the first few measurement modes of the federal 13-mode composite cycle, then are equal in the seventh or eighth modes, and finally are overstated by about 10 percent in the last modes⁽¹¹⁵⁾. These results and the study conducted by TRW suggest that the desiccant requires a period of time to equilibrate. If a new drier were used before each 13-mode test (permissible under present EPA regulations), NO levels would tend to be low (since levels would be understated until the drier had equilibrated, possibly not until the last, rated speed, low power modes were being measured). If

the drier was not replaced, or if it was preconditioned, the NO levels would tend to be overstated. This is probably the case for those large-bore engine manufacturers who use driers, since their engines are typically operated about an hour before they are stabilized and measurements are begun.

The EMD procedure is similar to the SAE procedure with the exception that both NO and NO₂ are measured using NDIR and NDUV analyzers. Results from the TRW study⁽¹¹⁶⁾, based on automotive exhausts, indicated that NDIR/NDUV NO_x measurements were about 20 percent higher than CL readings (no water removal for CL sample). Apparently, the positive biases (interference and drier) more than offset potential NO_x loss due to the cold sampling in that study. In addition to the positive NDIR biases, NDUV analyzers have a history of stability problems, usually manifested as a drifting (high) reading. One study attributed this error to interference due to carbon particle buildup⁽¹¹⁷⁾.

C.3.3.3 Summary and Conclusion of DEMA, SAE, and EMD Practices

In summary, it appears that the DEMA practices will generally lead to negative errors due to NO_x loss in transporting the sample from the engine to the analyzer. At this time the absolute uncertainty associated with this practice is unknown relative to the EPA procedure. Limited results (see Table C-87) suggest this error could be as much as -10 to -15 percent depending on the amount of NO₂ in the sample (estimated to be 2 to 10 percent of the total NO_x) and the extent that NO is converted to NO₂ during the sample transfer.

The positive bias of the SAE procedure gives a 12-percent error in the NO reading. The EMD procedure, which measures both NO and NO₂ causes a 20-percent error in the NO_x reading. Nevertheless, both procedures could

experience as much as a -40-percent error in the NO or NO_x reading if the chemical drier has not stabilized. Thus, the uncertainties in NO_x measurements for the SAE and EMD practices appear to be significantly greater than the DEMA practice.

Finally, the EPA procedure defined for the heavy-duty diesel and gasoline engines appears to have minimized the various sources of measurement error and should give more accurate and consistent NO_x readings. Therefore, each of the manufacturer's sampling practices will be evaluated in the following section by comparing it to the EPA procedure.

C.3.3.4 Comparison of Manufacturer's Measurement Uncertainty

Table C-91 summarizes each manufacturer's practice in terms of procedures that could lead to measurement error. Note that three of the four DEMA manufacturers using cold sample lines have located their water traps near the analyzer instead of at the engine as is recommended in the DEMA practice. Therefore, this setup will promote additional loss of NO_x in the exhaust sample. In addition, Colt and Waukesha have relatively high sample residence times which will also promote NO_x loss.

The potential errors in measurement resulting from these practices are estimated in Table C-92(118-122) relative to the EPA procedure. (Blanks appear in Table C-92 for items which do not apply to a particular manufacturer.) These uncertainties are depicted for each manufacturer in Figure C-22. Errors due to system leaks are not included in the figure since this error cannot be generalized. Also, the uncertainty bands were constructed assuming that errors of the same sign were additive. Note that

TABLE C-91. LARGE-BORE ENGINE MANUFACTURER'S SAMPLING PRACTICE

Practice Manufacturer	Sample Line			Water Removal			Data Recording	Leak Check	Converter Eff. Check (CL)
	Heated	Length, ft	Residence Time, sec	Refrig	Drier	Location			
<u>DEWA (CL)</u>									
Colt	No	20	14	Yes		Before Analyzer	Stripchart	Yes	2/yr
Cooper	No	20	3	Yes		Before Analyzer	Visual	No	No
Delaval	No	33	2	Yes		Before Analyzer	Visual	No	No
White Superior	Yes	30	4	Yes		Before Analyzer	Visual	Yes	?
<u>SAE (NDIR)</u>									
Maukesha	No	33	~15	Yes	Yes	Before Analyzer	Stripchart	No	--
Cooper	No	18-75	<3	Yes	Yes	Before Analyzer	Visual	No	--
<u>EMD (NDIR/NDUV)</u>									
GMC/EMD	No	40	6	Yes ^a	Yes ^a	Before Analyzer	Stripchart	Yes	--
<u>EPA (CL)</u>									
White/Alco	Yes	60-80	6-8	Yes		After Converter	Stripchart	Yes	Yes
Ingersoll-Rand	Yes	60	6	Yes		After Converter	Stripchart	Yes	Yes

^aEMD measures NO₂ with NDUV. No water traps are used.

TABLE C-92. MANUFACTURER'S SAMPLING ERROR AS A PERCENT, RELATIVE TO EPA PROCEDURE

68-1

Manufacturer's Practice	Sampling			Leak Check ^d	(121) Recording	NDIR ^e and Drier NDUV	Converter Check ^f
	Cold line, large residence time	Water trap before analyzer					
		Cold line ^b	Hot line				
<u>DEMA</u> Colt Cooper Delaval White Superior	-12 ^a	-5 -5 -5	<-5 ^c	-25 -25	±3 ±3		+ +
<u>SAE</u> Waukesha Cooper	-12 ^a	-5 -5		-25 -25	±3	$\begin{Bmatrix} -40 \\ +10 \end{Bmatrix}$ $\begin{Bmatrix} -40 \\ +10 \end{Bmatrix}$	
<u>EMD</u> GMC/EMD		-5				-40 +20	

^aPotential loss of NO_x based on Data from Schaub and Beightol (118).

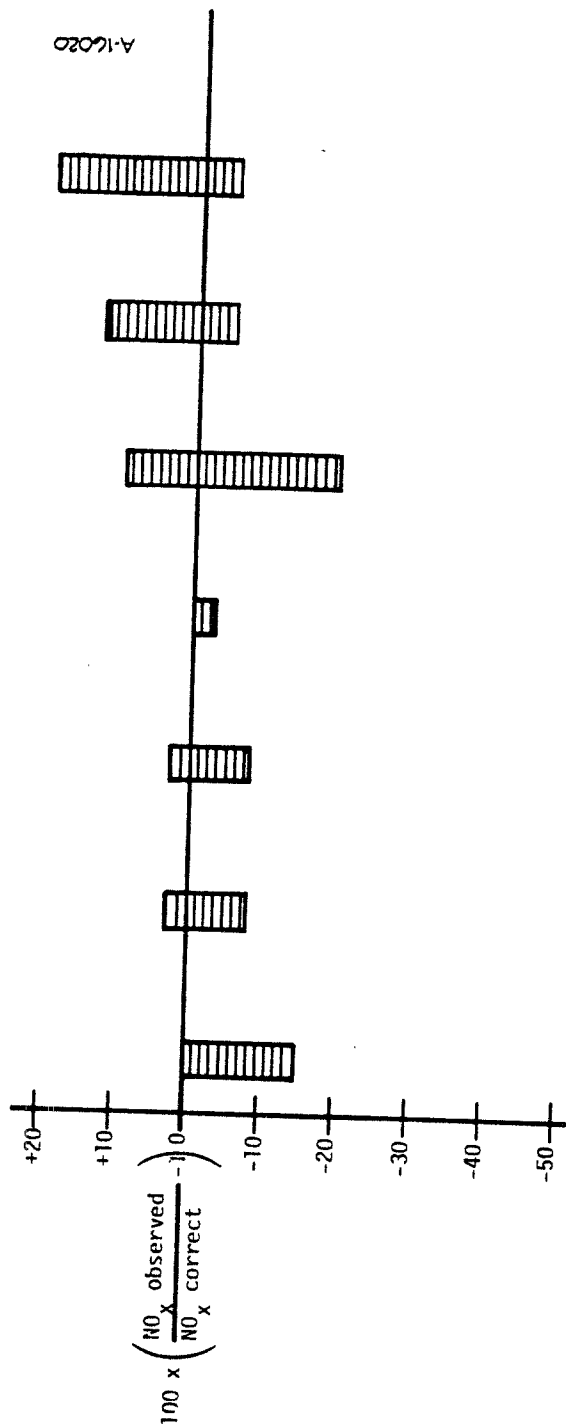
^bAssumes that 5 percent of NO converted to NO₂ and lost in water trap. Based on data from Clemmens (119).

^cLoss assumed to be less 5 percent since heated sample line used.

^dLoss cannot be generalized. Twenty-five percent loss based on one example (120).

^eForty percent error for unconditioned driers, 10 percent for NDIR (122).

^fFailure to perform converter checks could result in understated NO_x values due to incomplete conversion of NO₂ to NO.



Mfg	Colt	Cooper	Delaval	White Superior	Waukesha	Cooper	GMC/EMD
Practice			DEMA		SAE		EMD

^a Does not include potential error due to system leaks, and assumes chemical drier is conditioned.

Figure C-22. Measurement uncertainty relative to EPA procedure.^a

these uncertainties substantially exceed the 4-percent scatter in NO_x levels observed during verification tests of the EPA procedure.

NDIR and NDIR/NDUV instrument biases have been included in this figure and are based on the TRW study⁽¹²³⁾. Note that the potential negative 40-percent error of Table C-92 is not included in Figure C-22 since the manufacturers generally operate the engine long enough to condition the drier before measurements are taken. The data for Waukesha indicates that sampling errors can be as large as instrument bias, but in the case of Cooper and GMC/EMD, instrument biases predominate.

Conclusions

The above analyses of instruments and measurement practices suggest the following conclusions about the reported data:

- The exhaust data which were reported using the DEMA, SAE, or EMD practices should be banded by the appropriate uncertainty level. In general, measurement uncertainties are expected to range from -15 to +5 percent for DEMA data and -20 to +20 percent for the SAE/EMD data.
- Waukesha's data contains the greatest uncertainty since: (1) only NO was measured, (2) other significant sampling problems could have been present, and (3) the amount of NO₂ in the exhaust was unknown; thus correction of NO to NO + NO₂ would be speculative.
- The Ingersoll-Rand and White-Alco exhaust emissions data appear to have the least measurement uncertainty (±5 percent) since their sampling procedure was essentially identical to the EPA procedure.

The previous analyses also lead one to conclude that the EPA heavy-duty engine method should be preferred over any other method previously used for sampling stationary reciprocating engines. Therefore, any test method developed for these engines should follow the same basic sampling procedures and utilize the same basic sampling system. In the mean time, the DEMA and SAE/EMD sampling practices should include more carefully specified sampling procedures, and the use of NDIR/NDUU instruments should be carefully reviewed because of their bias relative to CL instruments.

C.4 THE EFFECT OF NO_x CONTROL TECHNIQUES ON NONMETHANE HYDROCARBON EMISSIONS

It is well known that the application of NO_x controls to IC engines can cause the emissions of other criteria pollutants to increase. Therefore, CO and HC emissions may require control as well as NO_x levels. However, attempts to regulate HC emissions from large-bore engines are not straightforward since most data for these engines are in terms of total hydrocarbon (THC), rather than reactive HC which is the criteria pollutant.

Moreover, the Office of General Council (OGC) has indicated that EPA should establish, if feasible, standards for the criteria pollutant, NMHC, to avoid legal challenges of a THC standard under Section 111 of the Clean Air Act(124). The OGC added, however, that EPA could set standards on THC if such an approach: (1) simplifies sampling procedures, and (2) allows the use of existing data. If EPA follows this approach, they must show that a standard for THC would cost manufacturers and users no more than an standard for NMHC. The purpose of this memo is to investigate this issue.

Data from Section 4.4 indicated that, in general, the application of NO_x controls to large-bore IC engines resulted in increases of 5 to 50 percent in THC levels (see Figure 4-52). However, the THC levels reported for the gas and dual-fuel engines contained both methane, (CH₄) a noncriteria pollutant, and NMHC, the regulated HC pollutant. (NMHC emissions are defined here as all other hydrocarbons in the exhaust sample.) If NMHC levels change differently than THC levels with the application of NO_x controls, separate control regulations may be required for NMHC levels. Therefore, the data for THC and NMHC (where measured) from gas and dual-fuel engines are reexamined here to determine how NO_x control techniques affect NMHC levels relative to THC levels.

In the following sections, uncontrolled levels of THC and NMHC will be compared for both new (laboratory) and installed (field engines). Then these emissions will be evaluated after the application of NO_x control techniques that were reported in Section 4.4. In addition, the measurement techniques for both NMHC and THC will be briefly reviewed. Based on this evaluation, a recommendation will be presented concerning the separate control of NMHC emissions.

C.4.1 Comparison of Controlled THC and NMHC Emissions from Gas and Dual-Fuel Engines

Previous investigators have shown that NMHC levels are generally less than 20 percent of the THC emissions from natural gas engines. Figure C-23 illustrates this conclusion using data reported by Southwest Research Institute (SwRI), and two engine manufacturers, Colt and Cooper. The SwRI data were from engines installed on gas transmission pipelines⁽¹²⁵⁾, whereas the manufacturer's data were from new engines that were measured in laboratories. All the THC data were measured with flame ionization detectors (FID's). Nonmethane emissions were determined as the difference in THC and methane levels. Cooper and Southwest measured methane emissions with a gas chromatograph. Colt physically removed all nonmethane portions of the sample and then measured the methane remaining with an FID.

Figure C-23 shows that THC levels for pipeline engines ranged from 1 to 9 g/hp-hr and were lower for four-cycle designs (1 to 3 g/hp-hr) compared to two-cycle levels (3 to 9 g/hp-hr). THC emissions from the engines tested by the two manufacturers were generally lower than those from the pipeline engines for both two- and four-cycle designs.

Although the THC levels for the new gas engines were lower than those for the installed engines, the NMHC levels were greater. As a consequence,

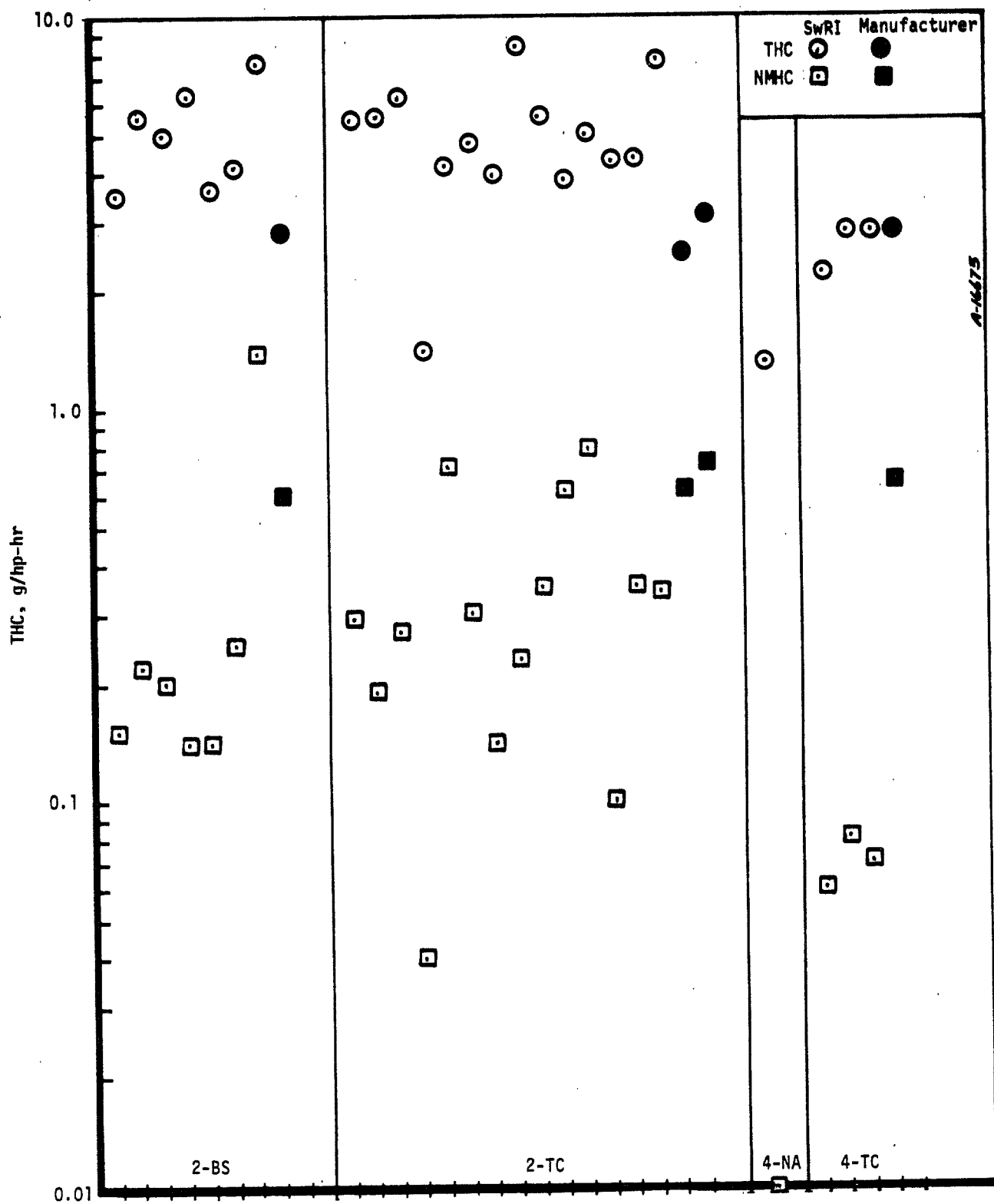


Figure C-23. Uncontrolled THC and NMHC levels from gas engines.

a greater fraction of the THC emission from the new engines were NMHC as compared to the pipeline units. On the average, NMHC emissions were only 6 percent of THC levels for the installed engines, but 23 percent for the new engines. These differences may be related to the condition of the installed engines which were tested as found. That is, older, installed gas engines may have had more blowby of unburned fuel and have been operating off peak combustion (i.e., requiring maintenance or an overhaul), thereby causing greater emissions of unburned fuel (methane). Furthermore, lower peak temperatures associated with less than optimum combustion may have resulted in fewer emissions of partially oxidized fuel (i.e., reactive hydrocarbons or NMHC).

Figure C-24 shows uncontrolled THC and NMHC levels for Colt and Cooper dual fuel engines. (No measurements of installed dual-fuel engines were available.) These data are similar to the limited data for manufacturer's gas engines, indicating that NMHC levels from these engines range from 10 to 20 percent of THC levels.

In the following sections, the levels of NMHC and the ratios of NMHC/THC for these gas and dual-fuel engines will be evaluated after the application of NO_x controls. These comparisons will show whether NMHC levels respond differently to controls than to THC levels, and, therefore, whether any standards would have to be written in terms of NMHC.

C.4.2 The Effect of NO_x Control Techniques on THC and NMHC Levels

Figures C-25 through C-30 illustrate how the application of NO_x control techniques affect THC and NMHC levels from gas and dual-fuel engines. Note that data were reported for both pipeline engines and those tested in manufacturers' laboratories. In general, these figures show that derate

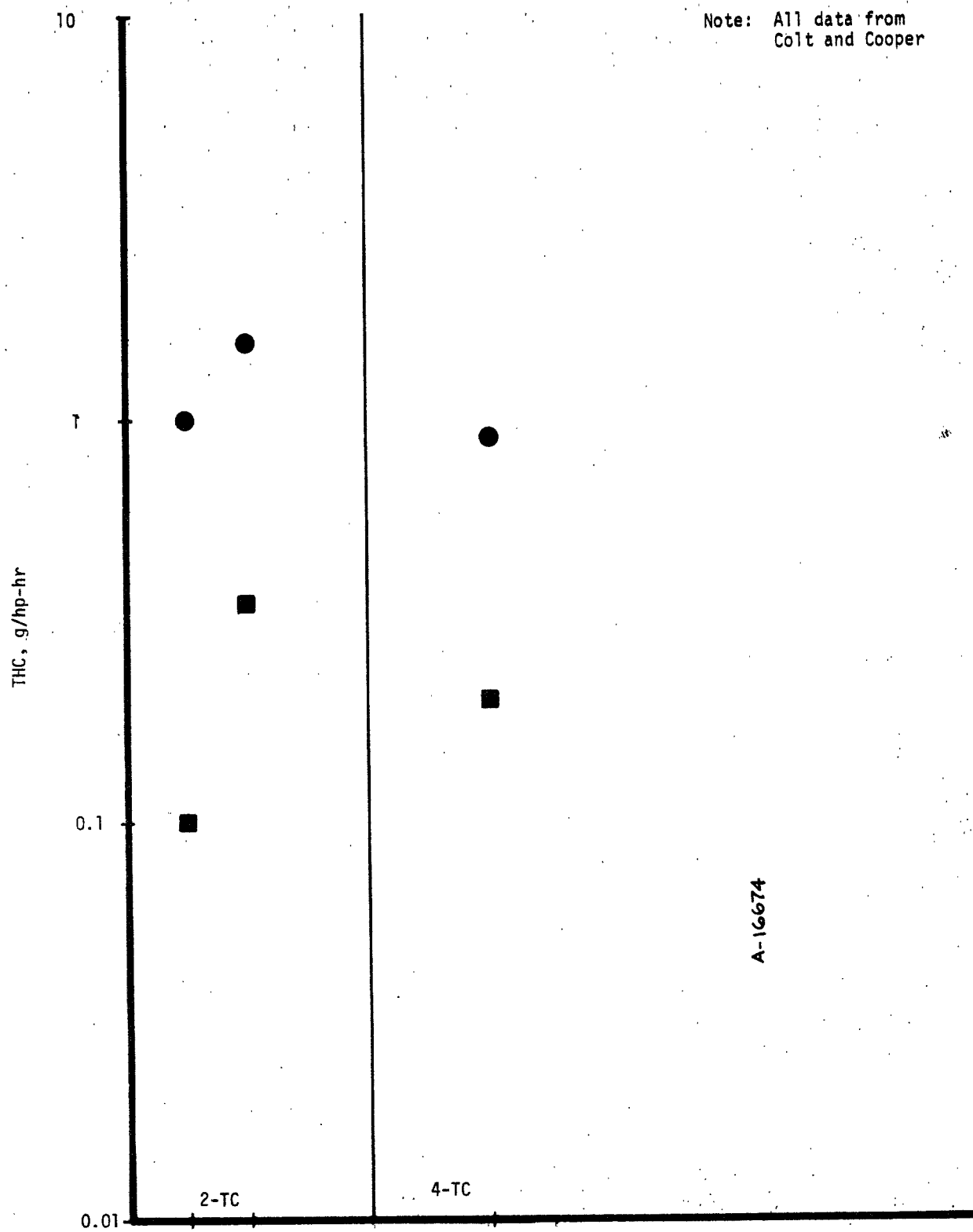


Figure C-24. Uncontrolled THC and NMHC levels for dual fuel engines.

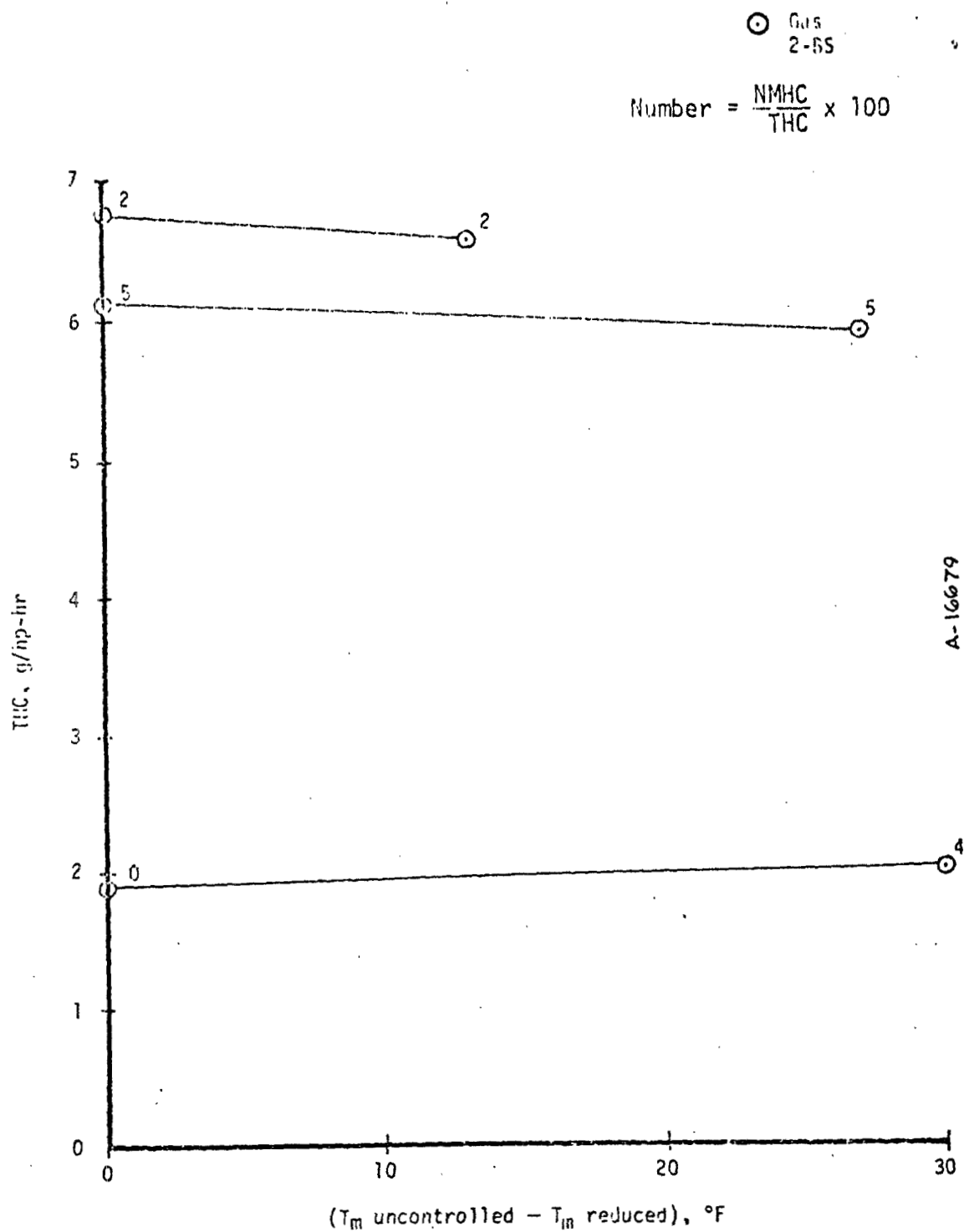


Figure C-25. Effect of derating on THC and NMHC levels.

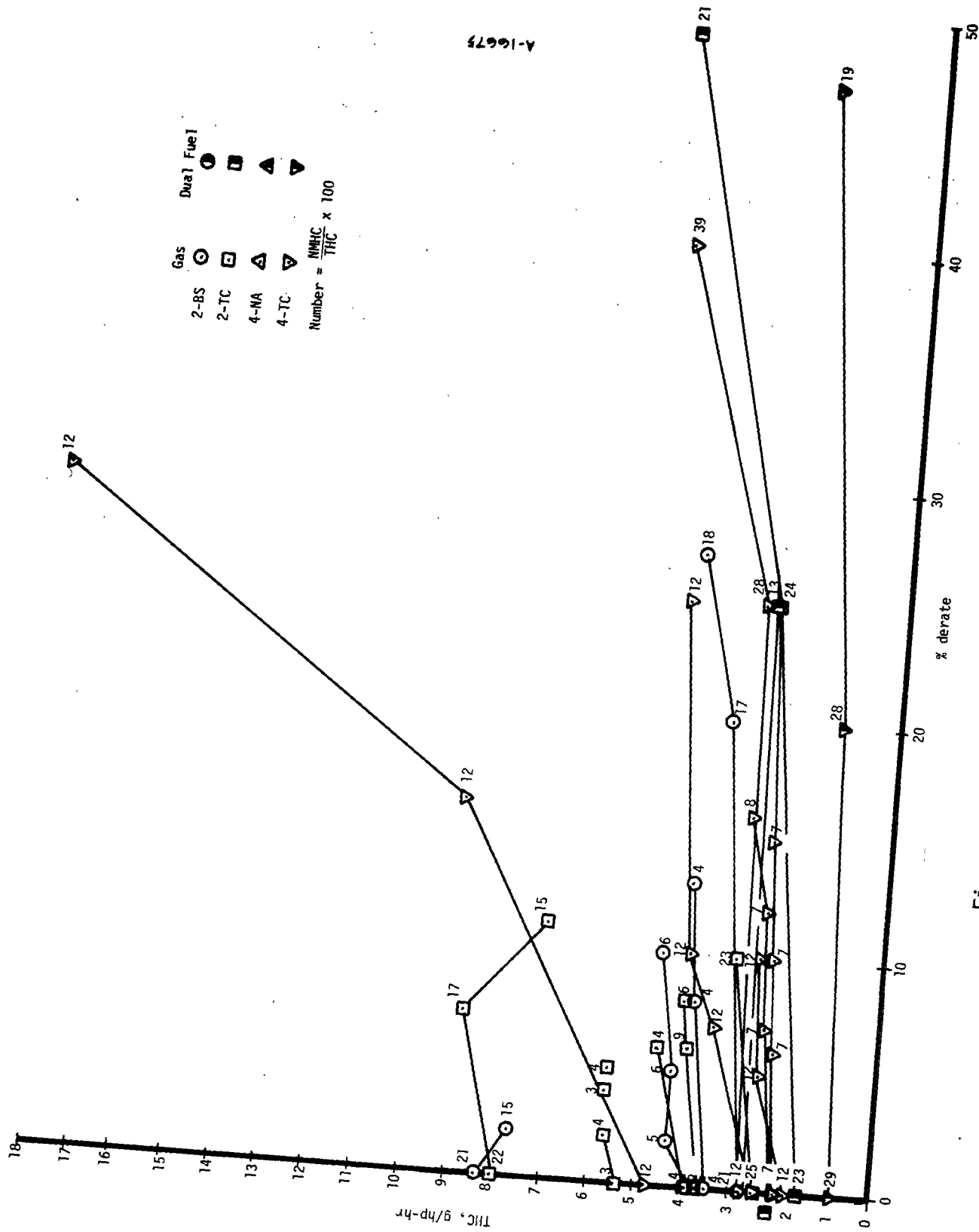


Figure C-26. Effect of A/F on THC and NMHC levels.

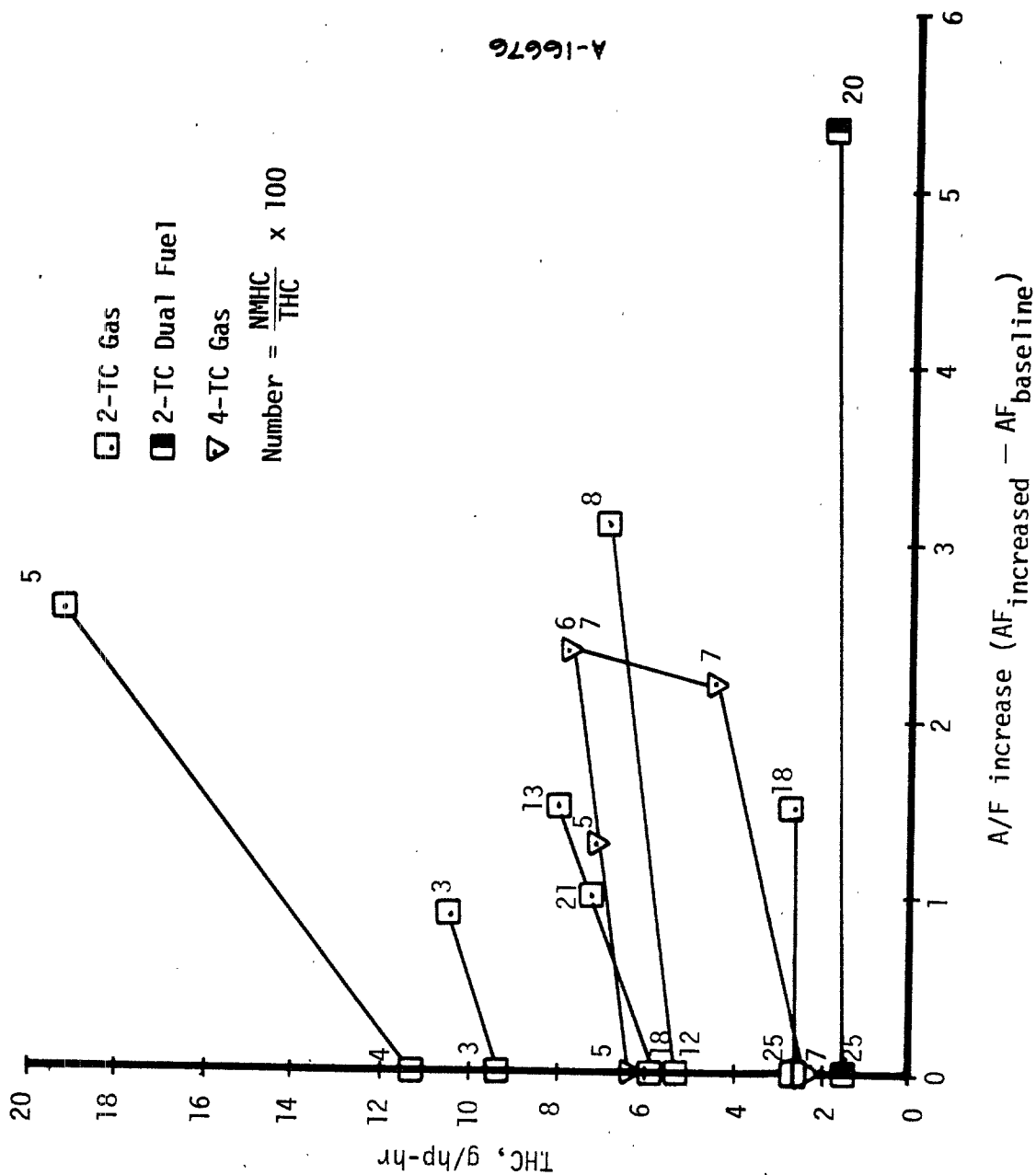


Figure C-27. Effect of retard on THC and NMHC levels.

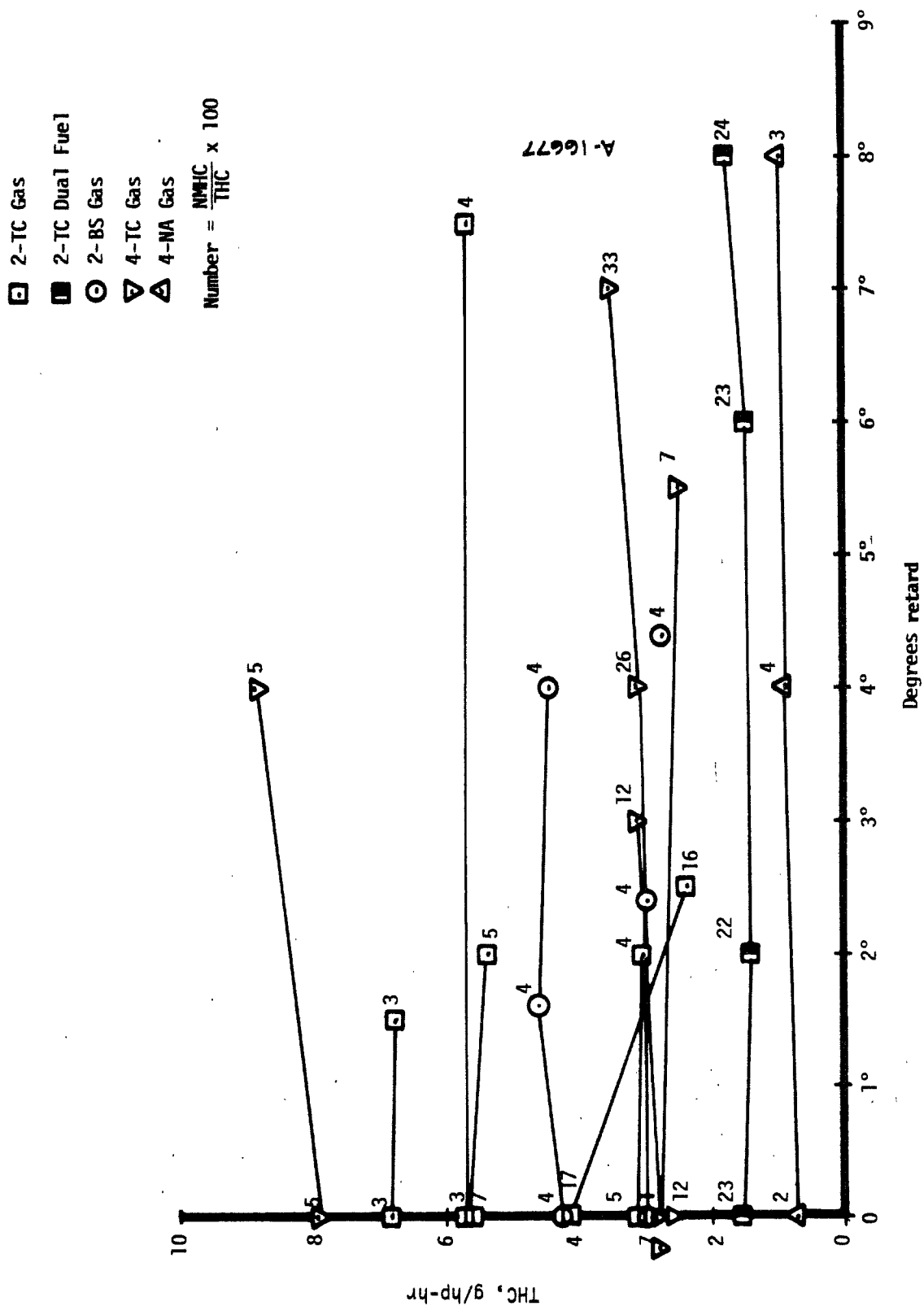


Figure C-28. Effect of EGR on THC and NMHC levels.

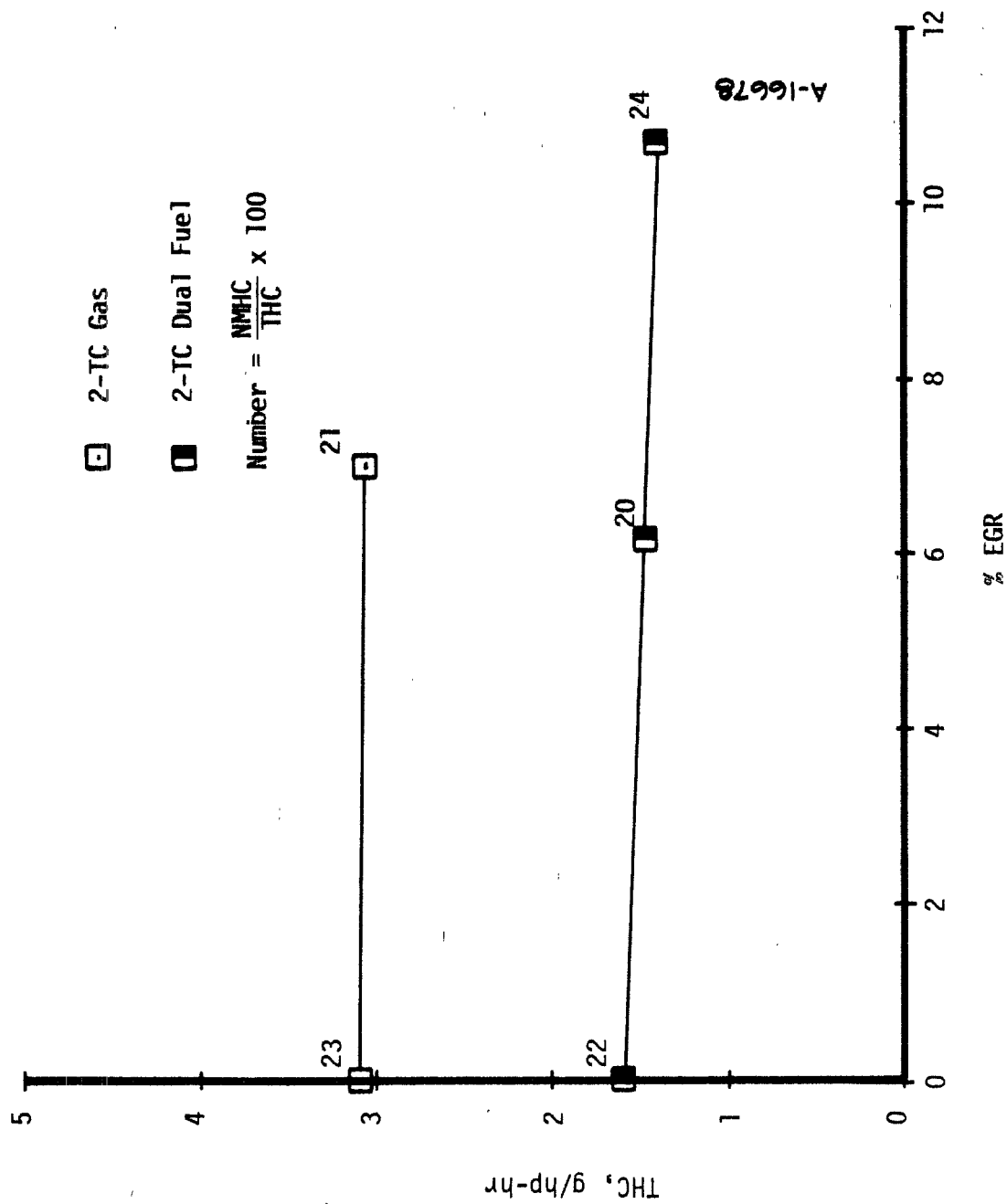


Figure C-29. Effect of manifold air cooling on THC and NMHC levels.

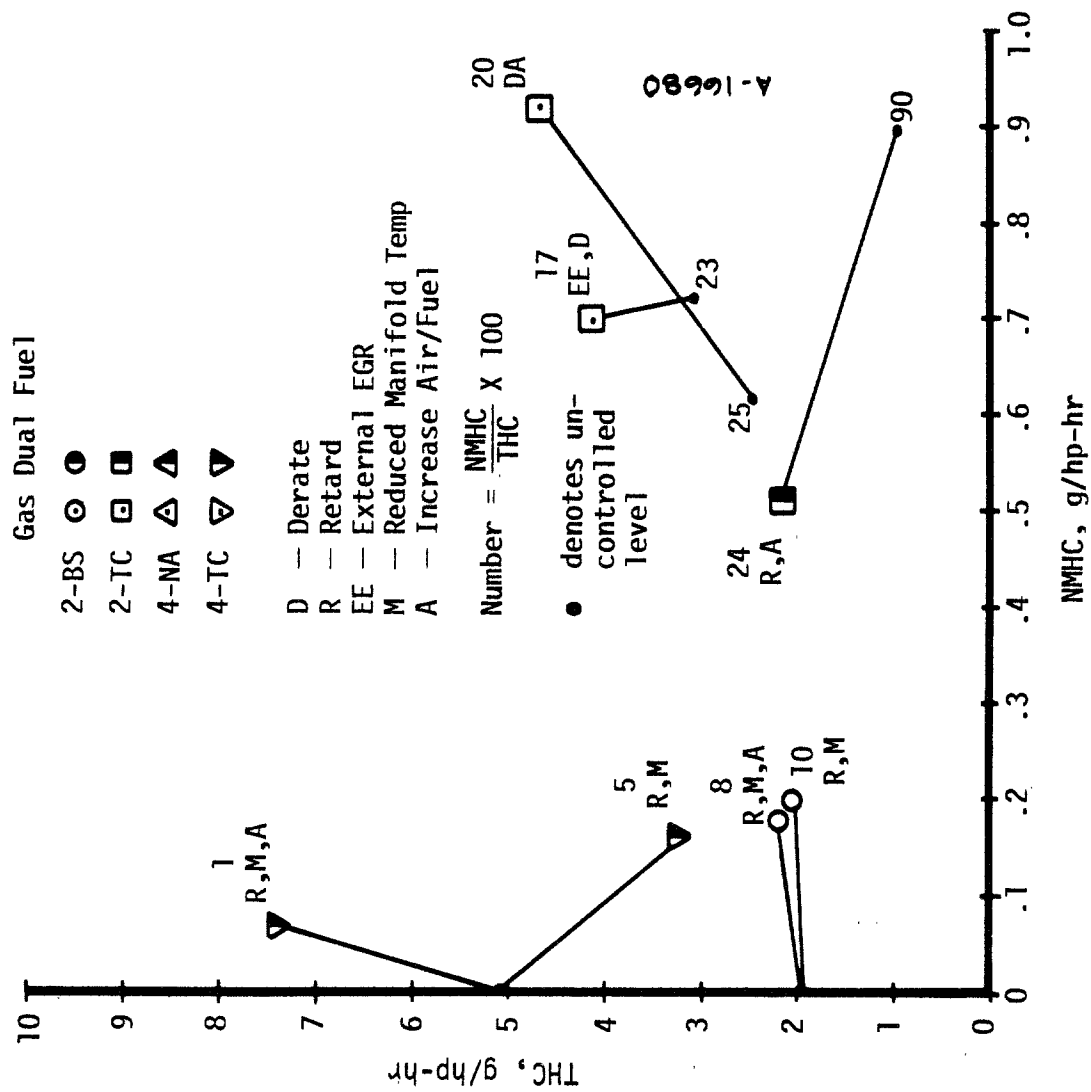


Figure C-30. Effect of combined NO_x controls on THC and NMHC levels.

and air-to-fuel ratio changes cause THC emissions to increase, but retard (R), external EGR, and manifold air temperature (MAT) reductions do not change the uncontrolled THC. Furthermore, for all of these techniques, NMHC levels remained about the same percentage of THC as they were in the uncontrolled engines. That is, NMHC emissions from both gas and dual-fuel engines change in the same proportion as do THC levels, or remain in the same proportion (where THC remained unchanged) with the application of NO_x controls.

Figure C-25 shows that, in general, derating caused THC to increase less than 50 percent, although extreme amounts of derate (>25 percent) caused uncontrolled THC levels to increase from 1 to 3 g/hp-hr to 2 to 5 g/hp-hr. In one case, THC emissions increased from 5 to 18 g/hp-hr for an installed pipeline engine that, possibly, required maintenance. In all cases, NMHC levels ranged from 5 to 25 percent of THC levels and did not change with derating. The increases in THC levels are probably related to the lower exhaust and peak temperatures of part load operation.

Similarly, Figure C-26 shows that increases in A/F cause moderate (10- to 25-percent) increases in THC levels. One 2-TC installed gas engine was the exception; THC levels increased 75 percent from 11 to 19 g/hp-hr, and may be related to a worn condition of the engine requiring maintenance (i.e., excessive blowby). Increases in A/F of gas engines that are already operating lean tend to lower peak combustion temperatures and prevent complete combustion. Consequently, HC emissions increase, although not as rapidly as with decreases in A/F from rich (less than stoichiometric) operation (see Figure 4-29). Note that the ratio of NMHC to THC remains essentially constant with increases in A/F.

Figures C-27 through C-29 indicate that retard, EGR, and MAT do not cause THC or NMHC levels to increase. The application of retard delays the combustion of the air/fuel mixture, resulting in higher exhaust temperatures which, combined with an excess of oxygen, effectively oxidizes most remaining unburned fuel. The application of this technique, however, causes fuel consumption to increase (see Figure 4-27).

In contrast to retard, EGR and MAT controls usually cause no increase in fuel consumption. It follows that if complete combustion is maintained (evidenced by unchanged fuel consumption) HC emissions should remain low or unchanged. With the application of EGR, unburned HC are recirculated or trapped and then combusted during the next cylinder firing. The limited data for EGR, illustrated in Figure C-28, show that NMHC and THC levels remain unchanged.

With the reduction of manifold air temperature, however, HC emissions can be expected to increase as the reaction of this specie proceeds at a lower temperature. Data shown in Figure 4-52 indicate that THC levels increase, but less than 25 percent, with manifold temperature reductions. Nevertheless, the limited data for both NMHC and THC levels shown in Figure C-29 indicate that these emissions remain unchanged.

Figure C-30 shows how the application of combinations of controls affects NMHC and THC emissions. The results of this figure are mixed. It should be noted that this data is limited, representing six engines from two manufacturers. For three of the six engines shown, combined controls cause THC and NMHC emission to increase in the same proportion. However, data for two 2-TC designs (one dual fuel, the other gas) indicated that NMHC emissions decreased (when THC levels increased) with the applications of: (1) retard

and air-to-fuel for the dual fuel unit, and (2) EGR and derate for the gas engine.

Opposing this result and of more concern, one 4-TC design showed that NMHC emissions increased from 0 to 0.2 g/hp-hr when THC emissions decreased from 5 to 3 g/hp-hr with the application of retard and MAT. Nevertheless, the NMHC level is small, representing only 5 percent of the THC level. It should also be noted that, despite precautions to avoid sample degradation, the manufacturer of this engine experienced some inconsistencies in determining NMHC levels (i.e., measured methane levels exceeded THC levels). Therefore, the following section will briefly consider the procedures for measuring both NMHC and THC before a recommendation concerning the separate control of NMHC emissions is presented in Section C.4.4.

C.4.3 Measurement Practices for NMHC and THC Emissions

The development of a separate standard for NMHC emissions is largely dependent on the demonstration of techniques for detecting NMHC directly, or those which detect NMHC emissions indirectly by measuring methane and THC. The NMHC level is then determined by the difference of the two measured levels. It should be noted that the accuracy of indirect techniques depends on the accurate determination of both methane and THC levels.

At the present time, there are basically three methods available for determining NMHC emissions. These are(126):

- Gas Chromatographs (GC): Provide direct measurement of methane by separation from other hydrocarbons. NMHC measurement is obtained indirectly by subtraction of methane from THC or directly by measurement of remaining nonmethane hydrocarbons. When NMHC is measured directly, the methane value is also obtained and can be

subtracted from a THC measurement (with FID) as a check of the NMHC measurement.

- Nondispersive Infrared (NDIR): Also provides direct measurement of methane. NMHC measurement is obtained indirectly by subtraction of methane measurement from independent THC measurement.
- Selective Combustor (SC): Gives a direct measurement of methane by selective combustion of all nonmethane hydrocarbon, allowing measurement of the remaining methane on a conventional THC analyzer such as a flame ionization detector (FID). NMHC measurement is obtained indirectly by subtraction of methane from THC.

All of these NMHC techniques use a conventional heated flame ionization detector (FID) to make one measurement of either NMHC using a GC, or THC using a GC, NDIR, or SC.

Gas chromatographs are commercially available and were used by Cooper(127) and SwRI to report NMHC levels from their engines. This method, however, cannot perform real-time analysis (continuous rather than bag or batch sampling), a capability desired by development engineers and present in other major emission instruments (e.g., NO_x , CO, SO_x analyzers). In addition, this instrument requires specialized training to operate and maintain.

Selective combustors have only recently become available commercially. They are simple and inexpensive compared to a GC but apparently require more development. This is because not all lighter paraffins (e.g., ethane and propane) are eliminated from the test sample during the selective removal of all NMHC emissions. This results in higher methane (or smaller NMHC) levels than were actually present. Colt used this method to report NMHC levels from

their engines. However, no comparisons with GC data were available to determine if this effect may have caused the Colt NMHC data to be understated.

The third NMHC instrument, the NDIR, exists as a prototype and also requires more development before becoming available commercially. Nevertheless, the instrument has demonstrated accuracy and freedom from interference for either batch or continuous sampling of methane emissions. It is also believed that it will require greater maintenance than existing NDIR's since it is more complex.

It should be noted that Fisher and Goodwin(128) supported the development of standards for NMHC's. They noted, however, that although measurements of NMHC levels were feasible, significant time and expense would be required to adequately develop the necessary instruments and procedures.

Cooperative studies of the SAE measurement method have also indicated that further development of heated FID's is required to reduce measurement uncertainties(129). The most recent study consisted of a comparison of data from one diesel engine emission source measured during round-robin tests. The results of this study showed that THC levels varied by 10 percent (on the average) for a given manufacturer and 22 percent among manufacturers. It was concluded that further development of this measurement method was necessary to reduce this scatter.

The preceding discussion indicates that measurement methods for NMHC have not been adequately demonstrated at this time. In addition, studies of FID instruments indicate that considerable measurement uncertainty can arise from these instruments even though an accepted practice is followed.

C.4.4 Recommendations for the Control of NMHC

Based on the preceding results, it is recommended that any control of NMHC levels be based on the measurement of THC levels. This can be supported by recognizing that: (1) in general, NMHC levels change in the same manner as THC levels with the application of NO_x control techniques; and (2) monitoring NMHC levels accurately is considerably more difficult than for THC since gas chromatographs or physical conditioning of the sample is required. Therefore, if standards for hydrocarbons are deemed necessary, they should be based on the reduction of THC, rather than NMHC, levels.

C.5 DEVELOPMENT OF CONVERSION $BSNO_x$ TO NO_x (ppm) CORRECTED TO 15 PERCENT O_2

All emissions data in Appendix C.1 had been reported in terms of brake specific emissions, $BSNO_x$ (g/hp-hr). In addition, applying percent emissions reduction to the sales weighted uncontrolled emissions levels (section 4.3.4) results in emission limits having the brake-specific NO_x format. Since a concentration format (ppm) has been selected for the standard, a method must be derived to convert the $BSNO_x$ limits to concentration limits.

C.5.1 Converting $BSNO_x$ to NO_x Concentration Format

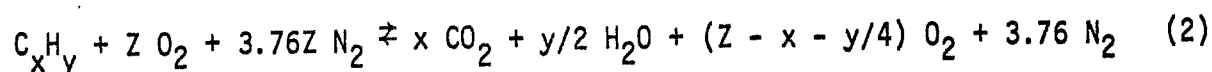
By definition, the volumetric concentration of NO_x in the exhaust is simply the volume of NO_x divided by the total exhaust volume. If the mass concentration is known, dividing each (the mass of NO_x and exhaust) by the respective molecular weights will give the volumetric concentration. This may be equally done on a brake-specific basis. Therefore, since the molecular weight of NO_x will be defined equal to NO_2 (46), the volumetric concentration of NO_x can be written as

$$NO_x \text{ (ppm) concentration} = \frac{\frac{BSNO_x}{46}}{\frac{\text{Brake specific exhaust gas, g/hp-hr}}{\text{molecular weight of exhaust gas}}} \quad (1)$$

Both brake-specific NO_x limits and the molecular weight of NO_x are known. However, the brake specific exhaust gas flow and the molecular weight of the exhaust gas are in general not available. Furthermore, to set emission limits for each fuel type, the average ratio of exhaust flow to exhaust gas molecular weight must be determined. This may be done by

assuming properties of the fuel and a reference 15 percent O_2 concentration in the exhaust gas. The following procedure has been used to determine the ratio of exhaust gas flow to exhaust gas molecular weight, using data obtained from available brake-specific fuel consumption.

For any hydrocarbon fuel, the chemical reaction may be written as:



This is a generalized equation, ignoring the small concentrations of other constituents such as NO_x , HC, CO etc., which are less than 1.0 percent. The moles of exhaust gas produced per mole of fuel burned is therefore $[x + y/2 + (Z - x - y/4) + 3.76 Z]$; and for dry products water is removed leaving $[x + (Z - x - y/4) + 3.76 Z]$ moles of dry exhaust. This reduces to $[4.76 Z - y/4]$ total moles of exhaust gas. The molecular weight of the gas is therefore the ratio of each constituent multiplied by the respective constituent molecular weight:

$$M_{\text{exhaust gas}} = \frac{x}{[4.76Z - y/4]} 44 + \frac{(Z - x - y/4)}{[4.76Z - y/4]} 32 + \frac{3.76Z}{[4.76Z - y/4]} 28 \quad (3)$$

or

$$M_{\text{exhaust gas}} = \frac{1}{[4.76Z - y/4]} (44x + 32Z - 32x - 8y + 105.28Z) \quad (3a)$$

reducing to

$$M_{\text{exhaust gas}} = \frac{1}{[4.76Z - y/4]} (12x - 8y + 137.28Z) \quad (4)$$

Since the standard assumes a 15 percent oxygen concentration in the dry exhaust gas, the moles of air, (Z), can be determined in terms of carbon and hydrogen in the fuel, (x and y), as follows:

$$0.15 = \frac{Z - x - y/4}{4.76Z - y/4} \quad (5)$$

$$0.714Z - 0.0375y = Z - x - 0.25y \quad (5a)$$

$$Z = \frac{x + 0.2125y}{0.286} \quad (5b)$$

$$Z = 3.5x + 0.743y \quad (6)$$

Therefore, by substitution into Equation (4),

$$M_{\text{exhaust gas corrected, 15\% O}_2} = \frac{492x + 94y}{16.64x + 3.29y} \quad (7)$$

The exhaust mass flow must be determined next. Since the total mass flow remains constant, the mass of exhaust gases per mole of fuel burned is:

$$\dot{m}_{\text{dry exhaust}} = \dot{m}_{\text{fuel}} + \dot{m}_{\text{air}} - \dot{m}_{\text{water formed}} \quad (8)$$

From equation (1) above, (Z) moles of air are burned with each mole of fuel, and (y/2) moles of water are formed. By multiplying by the molecular weight of each, the mass flows are as follows:

$$1 \times (12x + y) + Z[32 + 3.76(28)] - (y/2)(18) \quad (9)$$

Substituting equation (5) again (assuming 15 percent oxygen in exhaust);

$$\frac{\dot{m}_{\text{dry exhaust}}}{\text{mole fuel}} = (12x + y) + (3.5x + 0.743y)(32 + 3.76(28)) - (y/2)(18) \quad (10)$$

$$\frac{\dot{m}_{\text{dry exhaust}}}{\text{mole fuel}} = 492x + 94y \quad (11)$$

Dividing by the molecular weight of the fuel yields the mass of exhaust gases per mass of fuel, assuming 15 percent O_2 in the exhaust stream

$$\frac{\dot{m} \text{ dry exhaust}}{\dot{m} \text{ fuel}} = \frac{492x + 94y}{12x + y} \quad (12)$$

By multiplying this value by the brake-specific fuel consumption (BSFC) the brake-specific exhaust flow is determined.

$$\dot{m} \text{ dry exhaust} = \text{BSFC} \left(\frac{492x + 94y}{12x + y} \right) \quad (13)$$

The ratio of exhaust mass flow to exhaust gas molecular weight can now be determined. From equation (13) and (7):

$$\frac{\text{Brake specific exhaust flow}}{\text{Molecular wt. of exhaust gas}} = \frac{\text{BSFC} \times \left(\frac{492x + 94y}{12x + y} \right)}{\left(\frac{492x + 94y}{16.64x + 3.29y} \right)} \quad (14)$$

This reduces to

$$\text{BSFC} \times \left(\frac{16.64x + 3.29y}{12x + y} \right) \quad (15)$$

Substitution into equation (1) and converting to parts per million (ppm) gives:

$$NO_{x15} = \frac{BSNO_x \times 10^6}{(46) (BSFC) \left(\frac{16.64x + 3.29y}{12x + y} \right)} \quad (16)$$

where NO_{x15} is corrected to 15 percent O_2 in the exhaust.

To employ this equation, the values of x and y for the fuel must be known. In addition, BSFC must have the same units as $BSNO_x$ for substitution into equation (16). Thus, the following data for the fuel is required: 1) actual carbon and hydrogen makeup of the fuel C_xH_y or the hydrogen/carbon ratio y/x ; and 2) the fuel lower heating value Btu/g, since most fuel consumptions are expressed in Btu/hp-hr.

C.5.2 Determining Emission Limits

The above conversion method was used to establish emission limits in the following manner. First, equation (16) and the data in Appendix C.1 were used to generate plots of $BSNO_x$ and NO_{x15} for each fuel type. These are shown in Figures C-31, C-32 and C-33. For diesel and dual-fuel engines, diesel fuel with a hydrogen/carbon ratio of 1.8 and a LHV of 18,320 Btu/lbm was assumed. Natural gas was assumed to have a LHV of 20,000 Btu/lbm, and a hydrogen/carbon ratio of 3.5. The best fit line was then determined through all points as shown on the figures.

Once these best-fit lines were established, average BSFC corresponding to the lines were determined. These equal 7057 Btu/hp-hr for diesel, 7163 Btu/hp-hr for dual fuel, and 7609 Btu/hp-hr for gas engines. By averaging these values, an average BSFC of 7276 Btu/hp-hr was achieved, which represents an engine efficiency of 35 percent for all engines. This average value was placed into equation (16) and by using

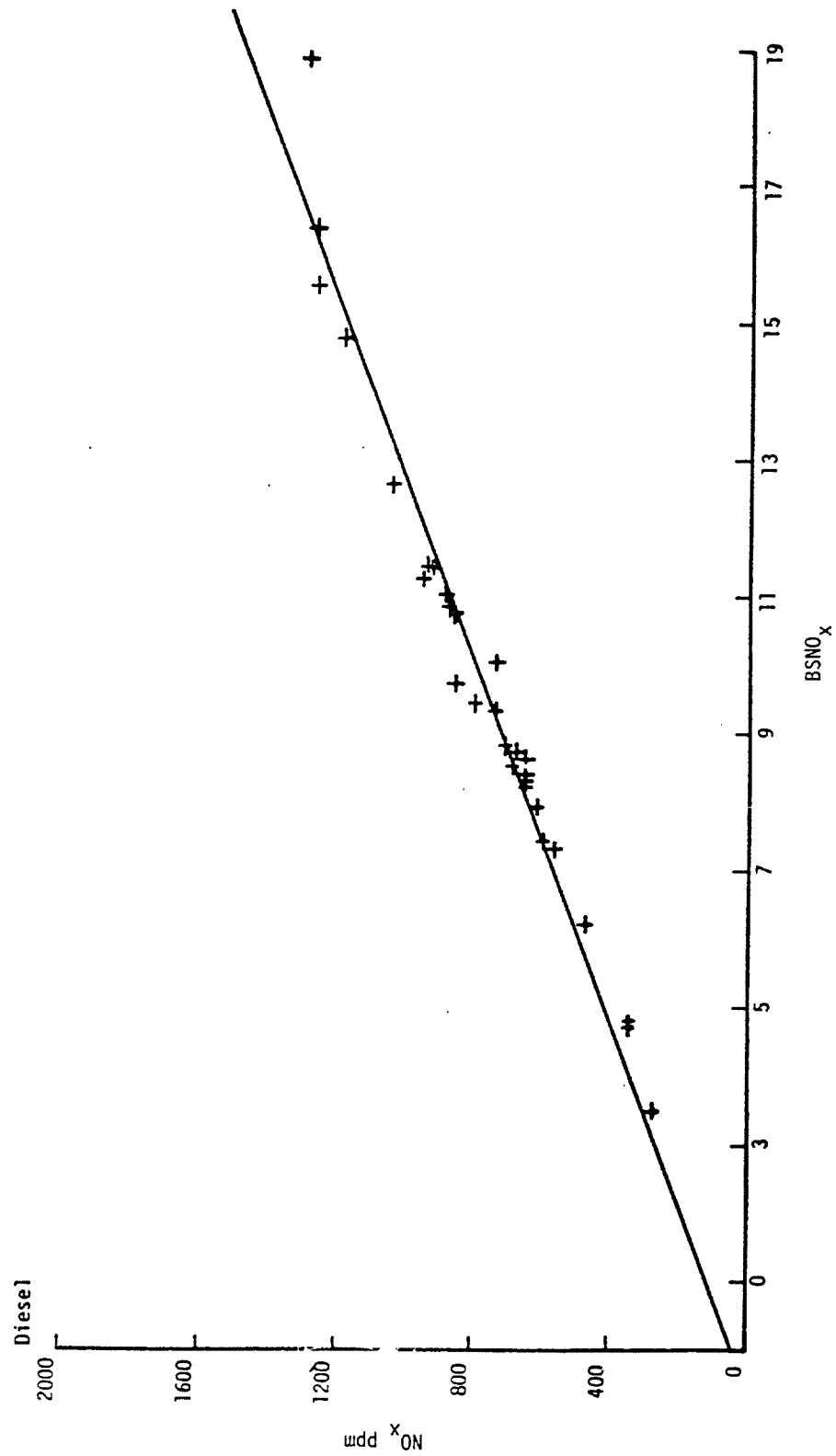


Figure C-31. BSNO_x converted to NO_x ppm; from existing uncontrolled diesel engine data.

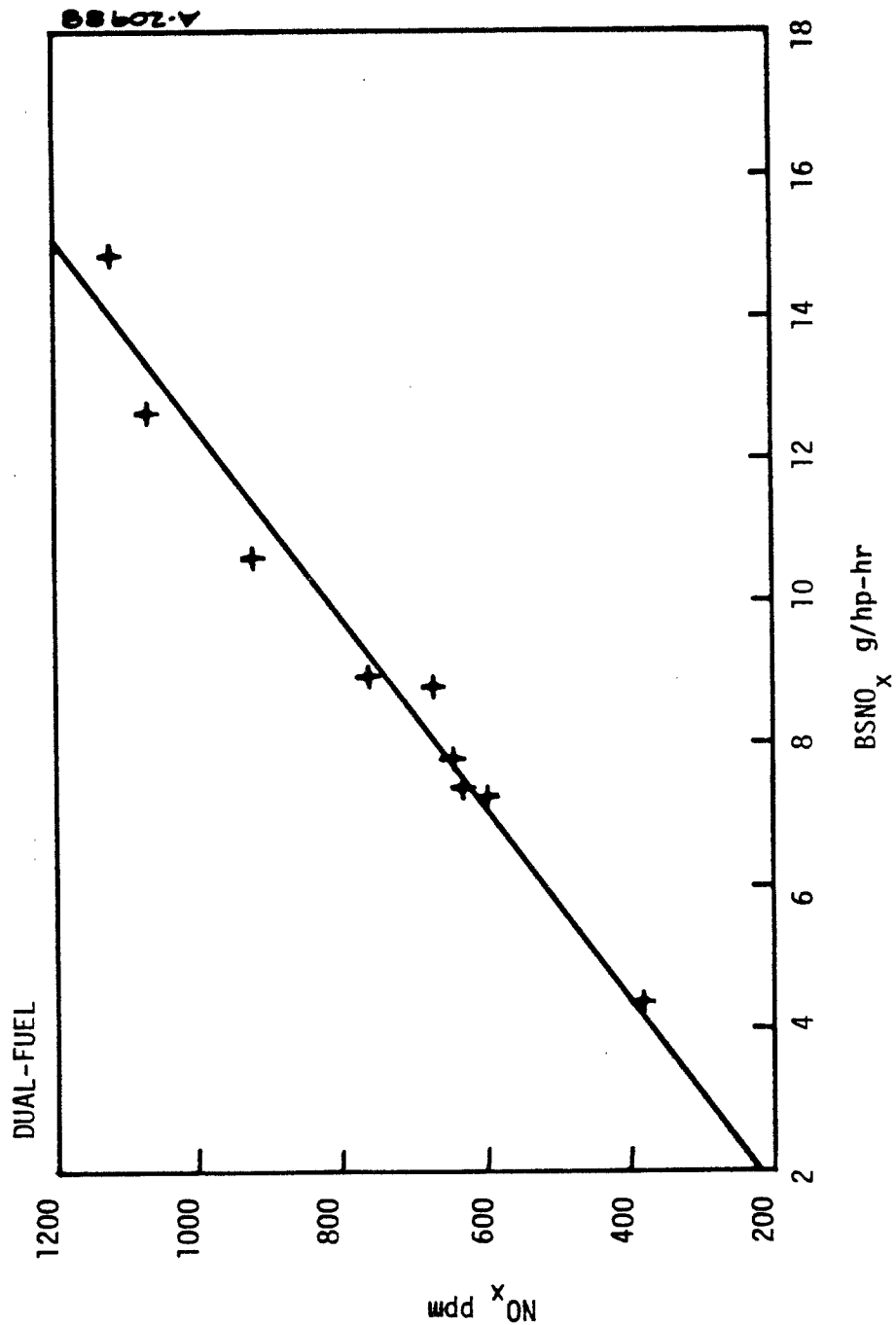


Figure C-32. BSNO_x converted to NO_x ppm; from existing uncontrolled dual-fuel engine data, assuming the engine being operated on 100-percent diesel fuel.

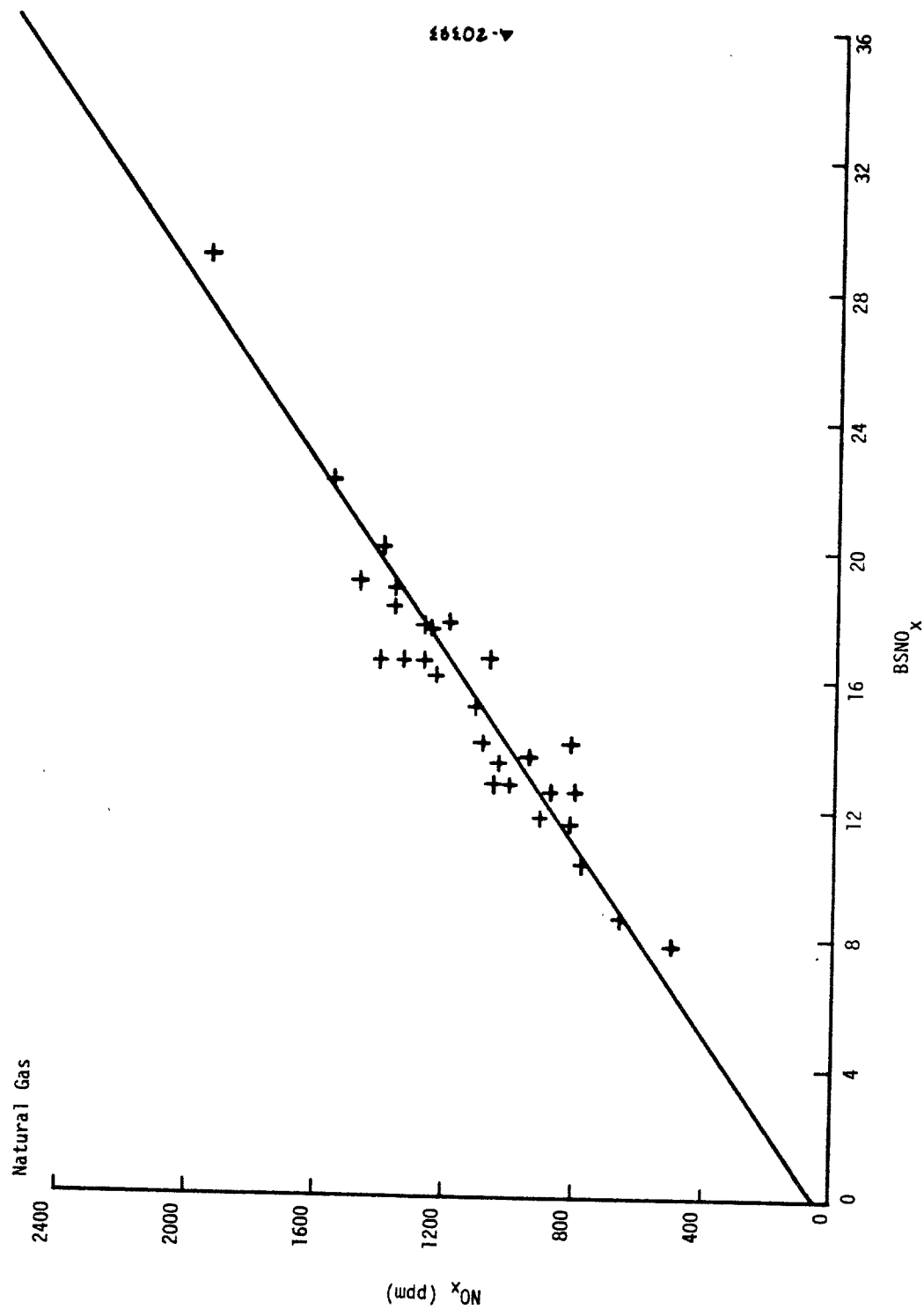


Figure C-33. BSNO_x converted to NO_x ppm; from uncontrolled natural gas engine data.

the limits of 5 g/hp-hr for dual fuel, 9 g/hp-hr for gas, and 7 g/hp-hr for diesel, (and the above fuel properties) the following NO_x emission limits were determined:

654 ppm for gas

518 ppm for diesel

370 ppm for dual-fuel

As a check of the validity of the conversion method, all raw ppm versus BSNO_x data was plotted (corrected to 15 percent O_2). By using equation (16), the average fuel consumption, and a mean value of the hydrogen/carbon ratio the average conversion curve was determined and is shown in Figure C-34. The standard deviation (expressed in percent) between this curve and the actual data was found to be 10 percent.

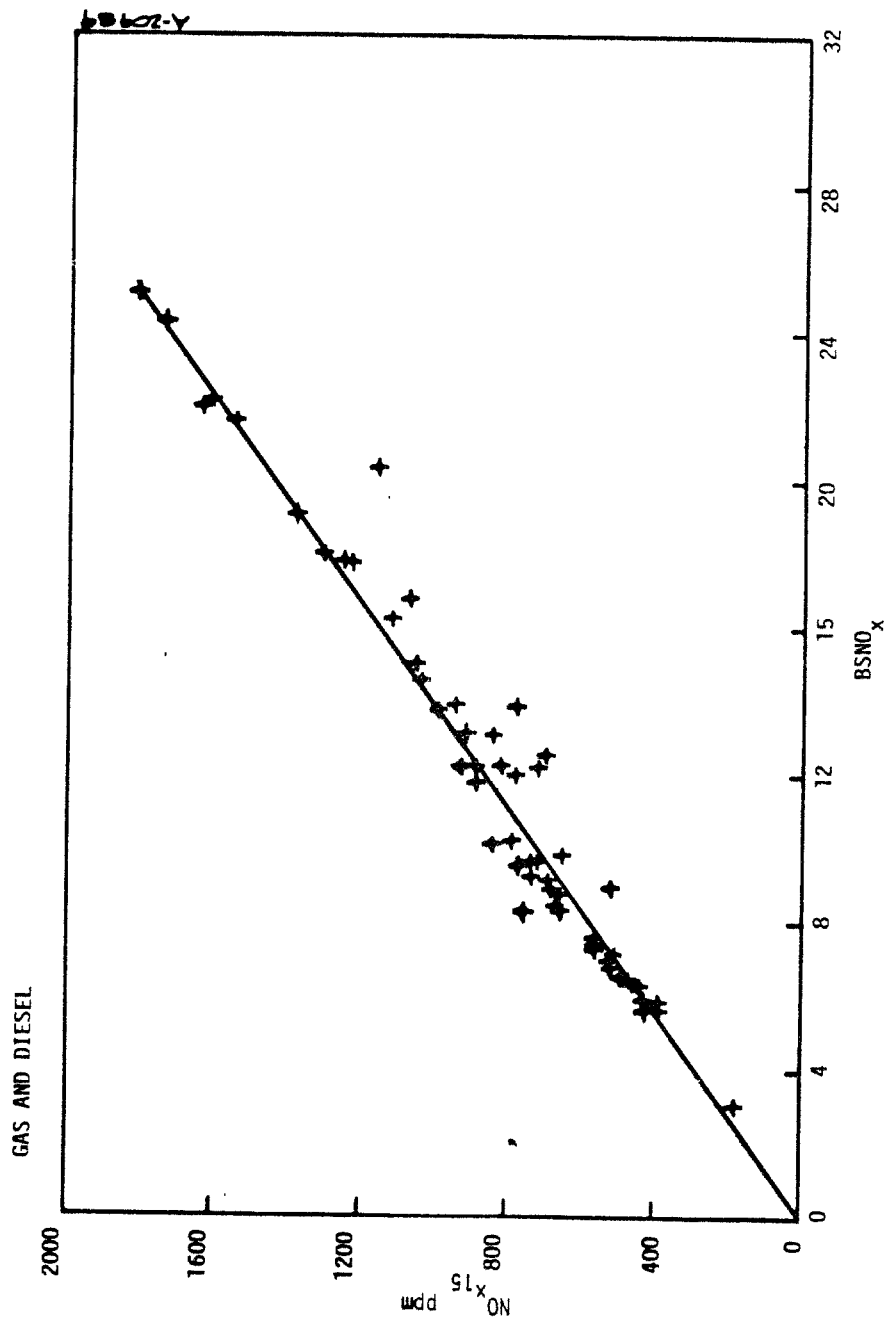


Figure C-34. Equation (16) plotted against raw diesel and gas ppm and BSNO_x data.

C.6 STATISTICAL ANALYSIS FOR ALTERNATIVE EMISSION LIMITS

In selecting the numerical emission limits for each fuel type, it was necessary to determine the tradeoffs associated with standards of performance based on each of the three alternatives of applying a 40 percent NO_x emissions reduction. Thus, a statistical approach was considered the most logical approach.

The uncontrolled NO_x emission from each engine fuel type were assumed to follow a theoretical normal distribution. Figures C-35, C-36, and C-37 illustrate the theoretical normal distribution curves for diesel, dual-fuel, and gas engines, respectively. Note that the curves have truncated ends at positive points on the axis as no engines emit 0 g/hp-hr. The areas under each curve represent the population if each engine fuel type emitted up to a particular limit. The mean values which bisect the normal distribution were assumed to be the sales-weighted uncontrolled average NO_x emission levels discussed in Section 4.3.4 and the standard deviations, for each fuel type were assumed to be equal to the standard deviation calculated from the data base in Appendix C. That is, the standard deviation is not sales weighted as no such information was available. If the standard deviation were calculated for increasingly large samples and increasing small class intervals, its value would be expected to approach the theoretical standard deviation. The data tabulated in Appendix C was used to calculate sample standard deviations for each fuel type: (1) Gas, $\sigma = 4$ g/hp-hr, (2) Dual-fuel, $\sigma = 3.2$ g/hp-hr, and (3) Diesel, $\sigma 3.7$ - g/hp-hr. As illustrated in the figures, about 68 percent of the engine population for a given fuel type will theoretically have an uncontrolled NO_x emission level within plus or minus one standard deviation of the mean.

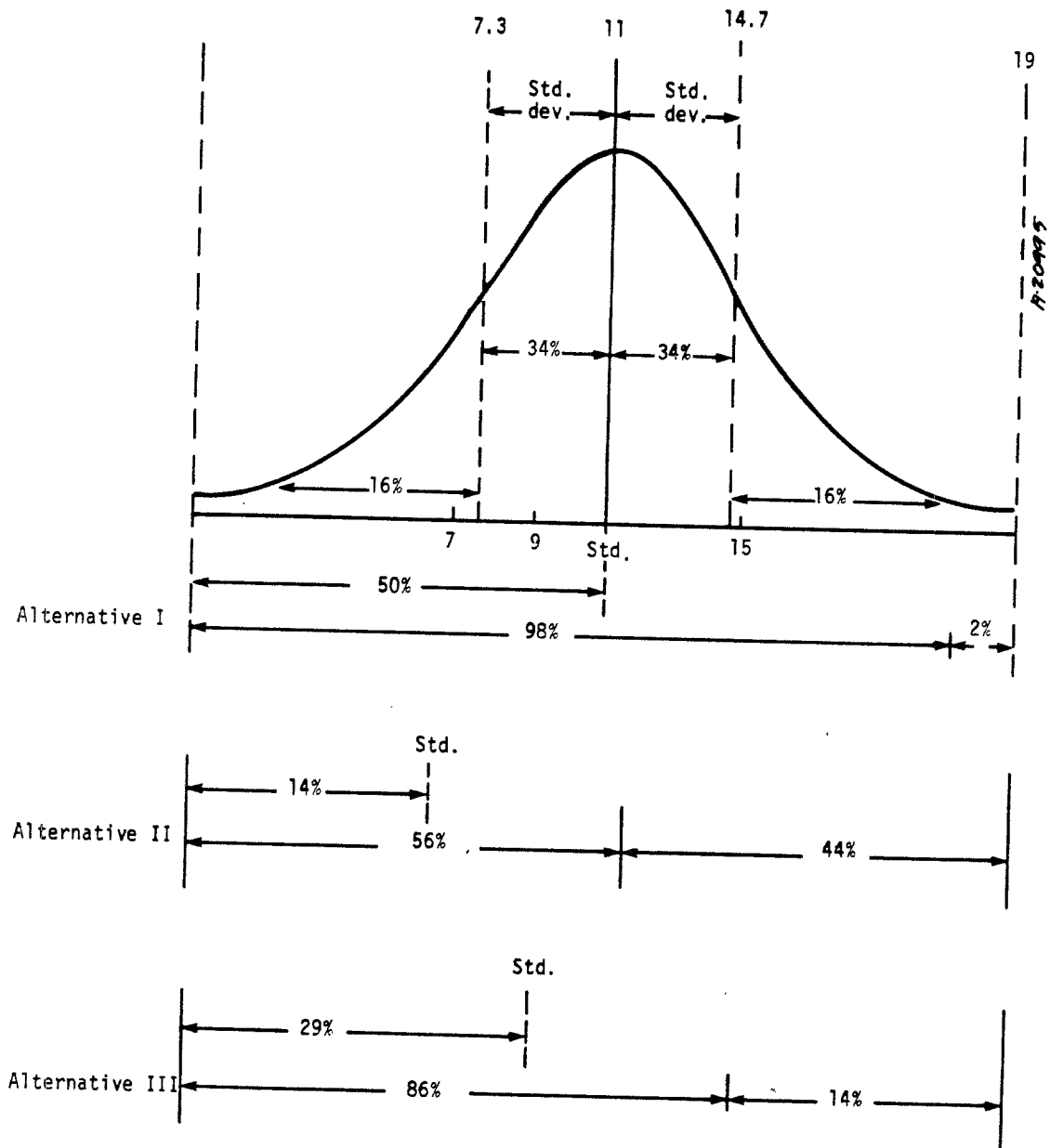


Figure C-35. Statistical effects of alternative emission limits on diesel engines.

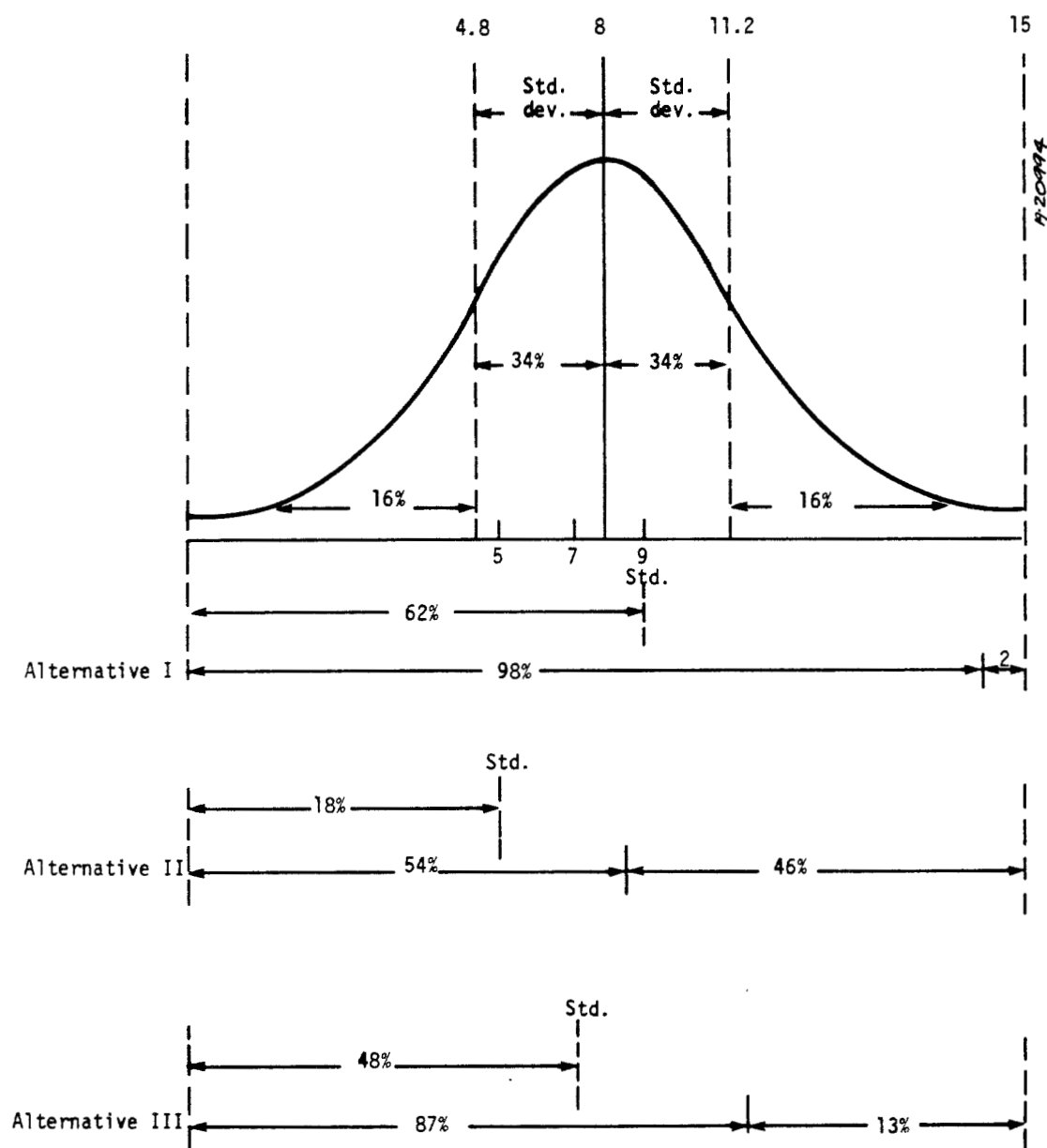


Figure C-36. Statistical effects of alternative emission limits on dual-fuel engines.

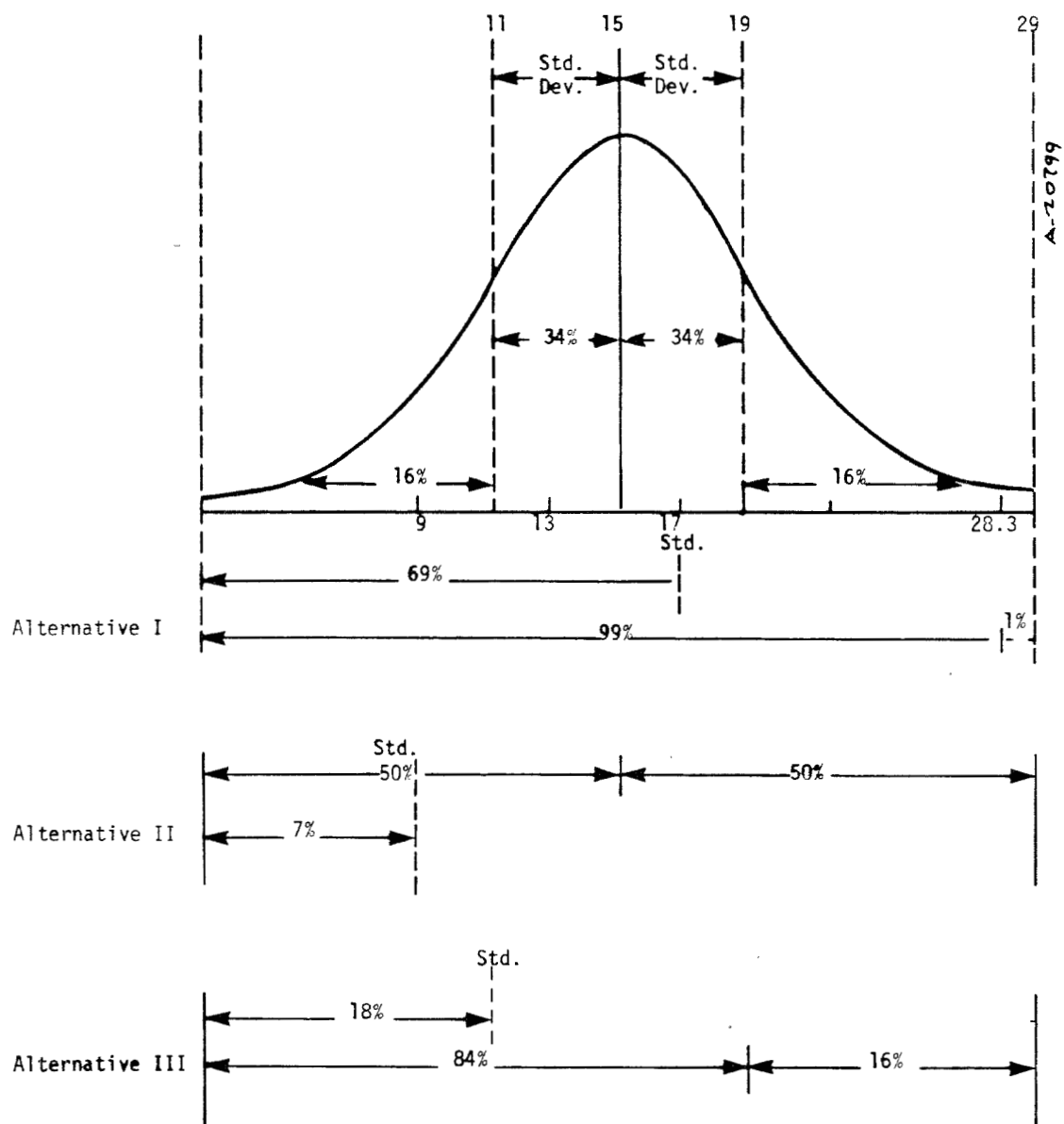


Figure C-37. Statistical effects of alternative emission limits on gas engines.

Standards of performance based on each of the three alternatives were analyzed relative to the theoretical normal distribution curves to determine: (1) the percentage of the engines that would have to reduce NO_x emissions by 40 percent or less to meet the standard; (2) the percentage of engines that would be required to do nothing to meet the standard; and (3) the percentage of engines that would be required to reduce NO_x emissions by more than 40 percent to meet the standard. The results are illustrated on each figure and summarized in Table C-93.

As a check on the accuracy of the assumption of a normal distribution, the actual data base in Appendix C was analyzed relative to percentages of engines for each alternative determined from the normal distribution. The results are tabulated in Table C-93 as the numbers in parentheses. The actual sample values show very good agreement with the values determined by the pure normal distribution approach.

Thus, it can be concluded that the assumption that the uncontrolled NO_x emission levels, for a given fuel type, follow a theoretical normal distribution curve is essentially true and the statistical approach is deemed to be a valid approach.

TABLE C-93. SUMMARY OF STATISTICAL ANALYSES OF ALTERNATE EMISSIONS LIMITS

GAS ENGINES			
Alternative	I	II	III
Standard	17	9	11
Percent meeting standard with ≤ 40 percent control	99 (100)	50 (50)	84 (91)
Percent required to do nothing to meet standard	69 (67)	7 (6)	18 (9)
Percent required to apply 40 percent control to	1 (0)	50 (58)	16 (9)
DUAL-FUEL ENGINES			
Alternative	I	II	III
Standard	9	5	7
Percent meeting standard with ≤ 40 percent control	98 (100)	54 (44)	87 (89)
Percent required to do nothing to meet standard	62 (67)	18 (4)	48 (44)
Percent required to apply 40 percent control to	2 (0)	46 (56)	13 (11)
DIESEL ENGINES			
Alternative	I	II	III
Standard	11	7	9
Percent meeting standard with ≤ 40 percent control	98 (100)	56 (63)	86 (90)
Percent required to do nothing to meet standard	50 (65)	14 (18)	29 (45)
Percent required to apply 40 percent control to	2 (0)	44 (37)	14 (10)

C.7 COMPARISON OF SMALL-BORE, AUTO ENGINE EMISSION CONTROL TECHNIQUES FOR LARGE BORE-STATIONARY IC ENGINES

C.7.1 Introduction

During a review meeting with EPA in September 1977, several questions were raised concerning the application to large-bore stationary engines of control techniques that have been shown to be effective on automotive engines. In addition, questions were asked why catalytic reduction of NO_x (which has been successfully used in Japan) and water induction (the primary NO_x control technique for stationary gas turbines) were not alternative controls for stationary IC engines. This memo briefly summarizes these three questions.

C.7.2.1 Comparison of Control Techniques for Large-Bore Stationary and Automotive Engines

Essentially the same NO_x control techniques that have been used by engine manufacturers to meet emission regulations for mobile sources are also effective when applied to large-bore engines. These techniques include derating*, retarded ignition or fuel injection, manifold air cooling (with turbocharging), air-to-fuel ratio changes, and exhaust gas recirculation. Of these controls, retard (for diesel or dual fuel engines) and air-to-fuel changes (for natural gas engines) are particularly effective when applied to large-bore engines.

Nevertheless, inherent differences in the design and operating modes of large stationary and smaller, mobile engines have dictated

*Derating is accomplished by using a larger engine than necessary for a particular car.

different approaches to exhaust emission control. Figure C-38, which shows the effect of A/F ratio on exhaust emissions, can be used to illustrate this point. Typically, spark ignition automotive engines are operated at slightly less than stoichiometric A/F ratios. This operation is largely dictated by the difficulty in closely controlling the A/F ratios to individual cylinders within these carbureted engines. Therefore, A/F ratios must be maintained richer than stoichiometric to avoid detonation and misfiring.

With the advent of strict emission regulations (primarily aimed at reducing HC and CO emissions), automotive manufacturers concentrated on increasing the A/F ratio (e.g., "lean burn" engines), which, as Figure C-35 illustrates, decreases HC and CO emissions significantly and also improves fuel economy. NO_x emissions, however, increase initially as A/F ratios are increased beyond stoichiometric. As emission regulations became even more stringent, additional controls were, therefore, required. These controls included exhaust gas recirculation (EGR) and returning to richer A/F ratios to reduce NO_x . To compensate for the resulting increase in HC and CO emissions, oxidizing catalysts were added. These catalysts also enabled automobiles to reach the more stringent requirements for HC and CO that became mandatory in 1975. Improvements in carburetion and mixing were also effective in optimizing exhaust emissions over a range of loads and speeds.

Large bore engines, in contrast, typically operate leaner than stoichiometric. Consequently, as shown in Figure C-35, NO_x emissions

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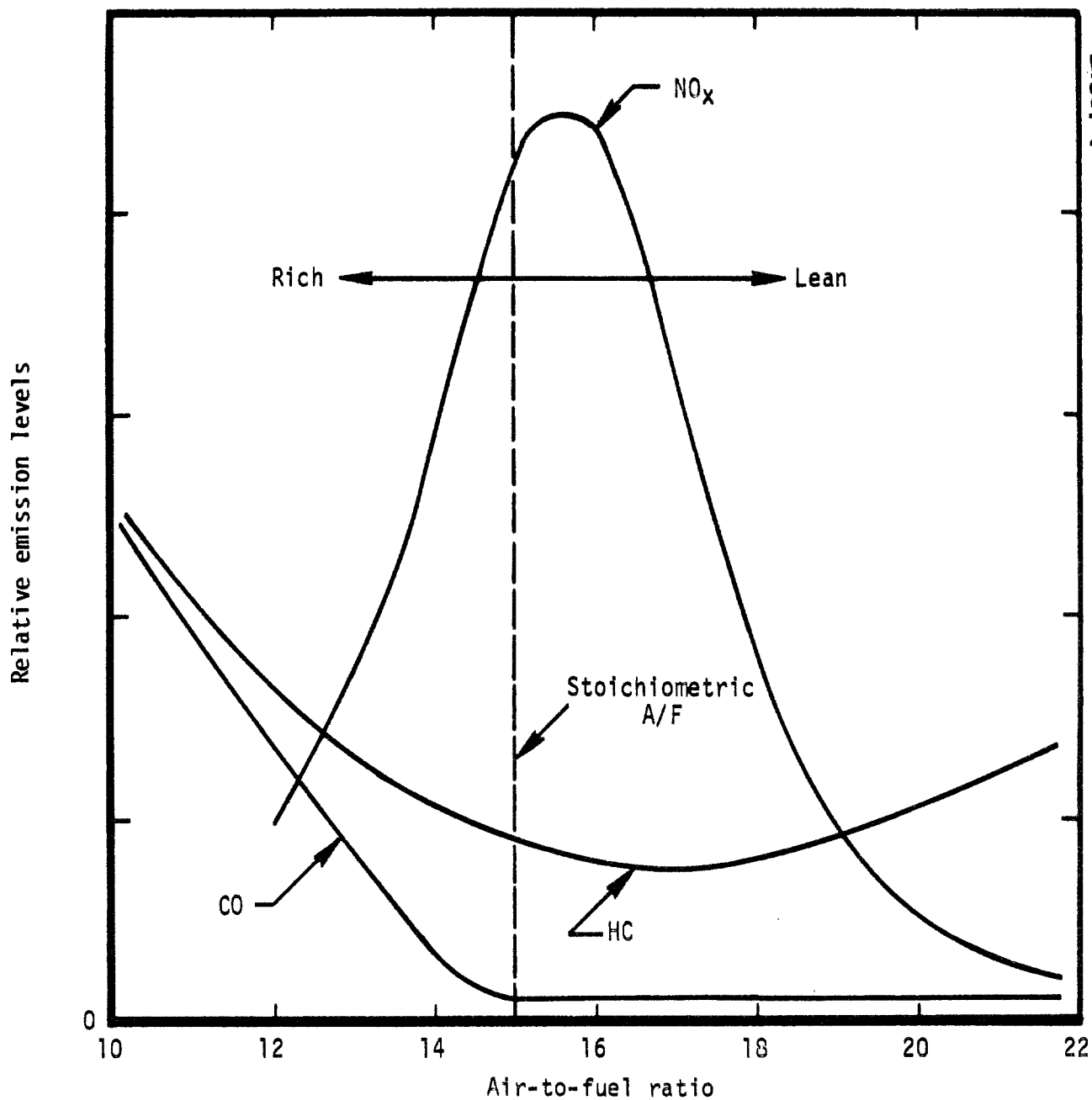


Figure C-38. Effect of A/F ratio on emissions from a gasoline engine.

are relatively high, but HC and CO emissions are now low*. These operating conditions result from the need to optimize fuel consumption of large-bore engines because they see high usage, high load service. Therefore, the emphasis of exhaust emission control for large-bore engines is in reducing NO_x emissions. Although the same types of combustion modifications that are effective in reducing NO_x from automotive engines could be used on large stationary engines, the selection of optimum controls and the direction and degree of application of the selected controls is different due to differences in uncontrolled A/F ratios and fuel charging methods.

C.7.2.2 Catalytic Reduction

As of this date no large-bore engine manufacturers are using catalysts to reduce NO_x emissions. NO_x reduction catalysts, until 1977, have not been used on gasoline or diesel vehicles either. The catalysts that have been and are still currently in use on automobiles are primarily oxidizing catalysts, whose purpose is to lower HC and CO emissions. Recently, however, three-way catalysts have been developed for the reduction of HC, CO, and NO_x emissions and the first commercial application was for a 1977 Volvo sold in California. (Three-Way Conversion Catalysts -- Part of the New Emission Control System; SAE Paper, 770365). These three-way catalysts are similar to the earlier oxidizing catalysts (precious metals coated on monoliths or pellets) with

*The exceptions to this generalization are those large-bore engines which are naturally aspirated or carbureted; these units have emissions characteristics which are similar to automotive engines since they operate at A/F ratios closer to stoichiometric.

the addition of rhodium which selectively reduces NO_x when the A/F ratio is maintained very near stoichiometric. These conditions exist in automotive exhausts but not in the discharge from large bore engines. Therefore, this control approach shows little promise for large bore engines.

As discussed on page 4-139 of the SSEIS, however, an ammonia/catalyst NO_x control system has been suggested in the literature and is based on the successful application of this technique to nitric acid plant tail gases and a number of large stationary combustion sources in Japan. An excellent discussion of technique is presented in a paper "Status of Flue Gas Treatment Technology for Control of NO_x and Simultaneous Control of SO_x and NO_x ", Mobley, J. David, and Stern, Richard D., U.S. Environmental Protection Agency, Report No. EPA-600/7-77-033c. This paper is an excellent review of this technology, its effectiveness, and its costs. Based on information in this report it would appear that selective catalytic reduction is a technically feasible approach for gaseous fuel engines, but the fouling (particulate) and catalyst-poisoning (sulfur) problems associated with oil or coal combustion would require further development. Moreover, this approach appears to be very expensive. If the published costs are extrapolated for IC engines, the data indicate that annual ownership and operating costs would increase from between 3 to 30 percent. These costs are based on sources that are at least an order of magnitude larger than a typical engine installation. In addition, the costs of maintenance and catalyst replacement may be somewhat understated for smaller sources, such as IC engines. Thus, it is probable that the actual costs for an engine application would be even greater than these estimates.

A similar NO_x reduction scheme has been developed by Exxon (Thermal Denox Process) for NO_x reduction from stationary boilers and furnaces. In this process, ammonia is injected into the flue gas at locations where the temperature exceeds 1500°F. There appears to be no application for this process to IC engine exhausts, however, since exhaust temperatures are generally less than 1000°F.

C.7.2.3 Water Induction

As discussed in the SSEIS (Section 4.4.7), water induction has been investigated by four large bore engine manufacturers (White Superior, Ingersoll-Rand, Cooper, and GMC/Electromotive). All reported serious concern about the feasibility of this technique based on observations of water in the crankcase and lubricating oil, rapid build-up of mineral scale (untreated water) around intake/exhaust valves and other components, and combustion deposits. All of the tests were of short duration (less than 25 hours of engine operation), and manufacturers believe much longer tests (2000 to 8000 hours) would be necessary to establish the effect of water induction on wear rates and operating reliability.

In addition to these manufacturers, two smaller bore engine manufacturers, GMC/Detroit Diesel and Caterpillar, have reported tests of water induction in truck-size diesel engines. Caterpillar believes the technique is viable if the water is treated for freeze protection and mineral content. Caterpillar tests showed significant deposits, but they concluded these could be prevented by demineralizing the water. GMC/Detroit Diesel also experienced significant deposits and concluded that these deposits originated from condensation of combustion products rather than water minerals. Based on their experience, GMC/Detroit Diesel concluded that water induction was not feasible in their 2-stroke designs.

C.8 Determination of Sales-Weighted Average Uncontrolled CO and HC Emissions

Since the source of variability caused by engine design cannot be specifically identified, a procedure is required to characterize uncontrolled CO and HC emissions levels of engines which are sold for similar applications.

The procedure adopted here is to compute a weighted, average uncontrolled emission level for engines in the diesel, dual-fuel, or natural gas categories. The three weighted levels are based on sales of engine horsepower during the past five years for domestic applications. Sales of horsepower to standby services were excluded from this computation, since engines sold for standby applications will be exempted from standards of performance (see Chapter 9).

The sales-weighted averages for diesel, dual-fuel, and natural gas engines are presented in Figures C-39(a-f), which also show each manufacturer's CO and HC_t data. The weighted average uncontrolled CO level for diesel engines is 2.9 g/hp-hr; for dual-fuel units, 2.7 g/hp-hr; and for natural gas engines, 7.7 g/hp-hr. The weighted average uncontrolled HC_t level for diesel engines is 0.3 g/hp-hr; for dual-fuel units, 2.8 g/hp-hr; and for natural gas engines, 1.8 g/hp-hr.

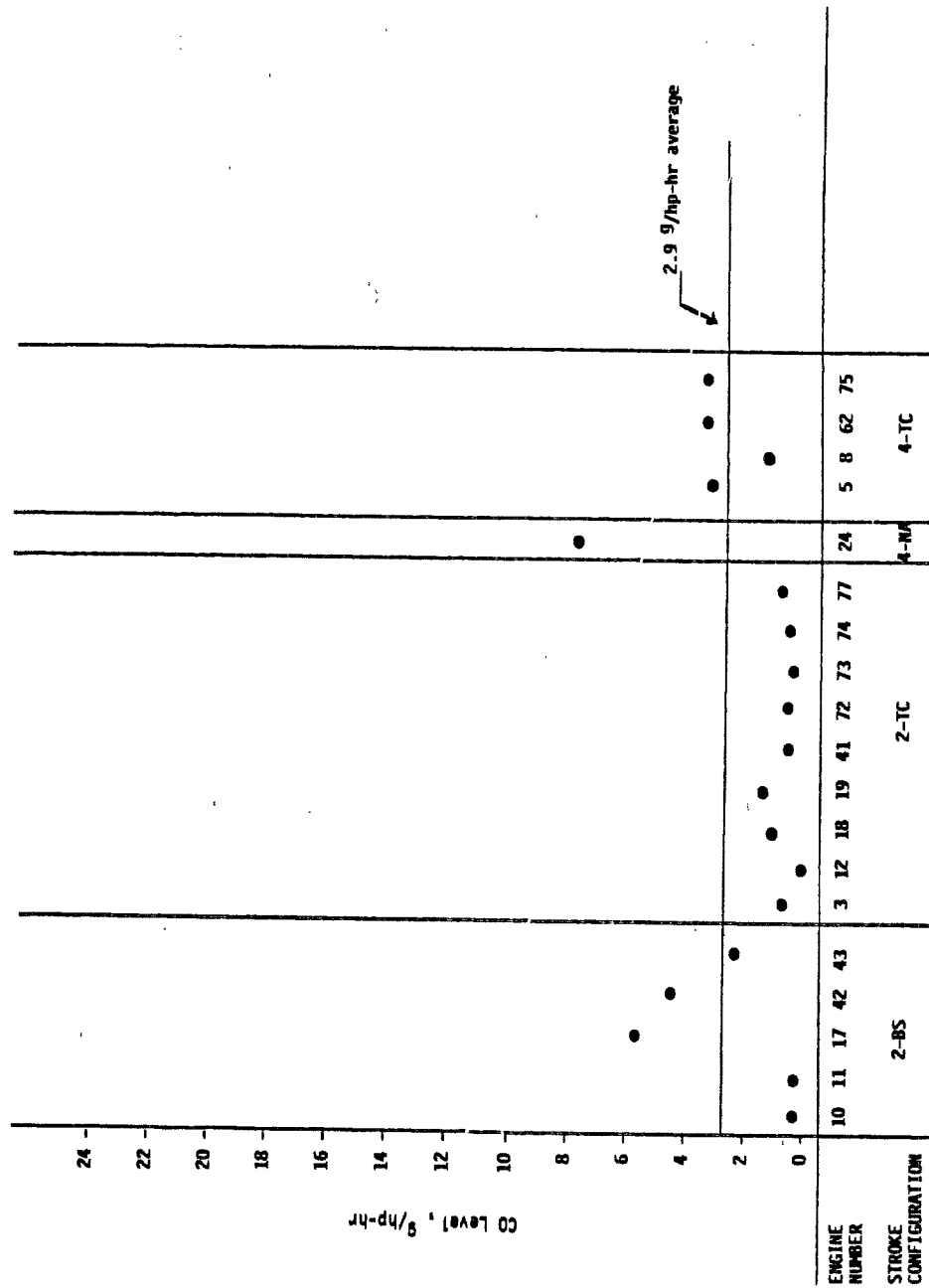
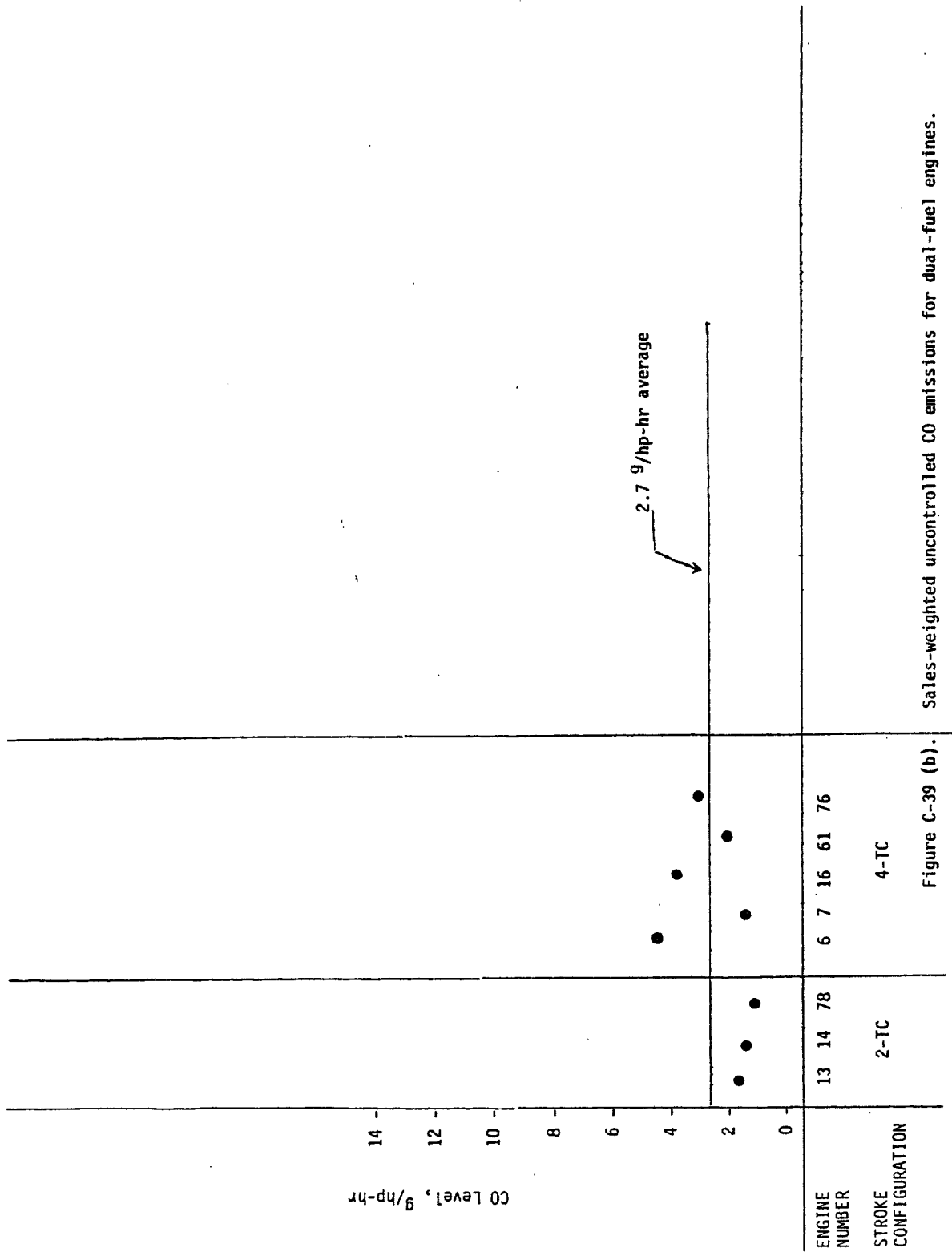


Figure C-39 (a). Sales-weighted uncontrolled CO emissions for diesel engines.



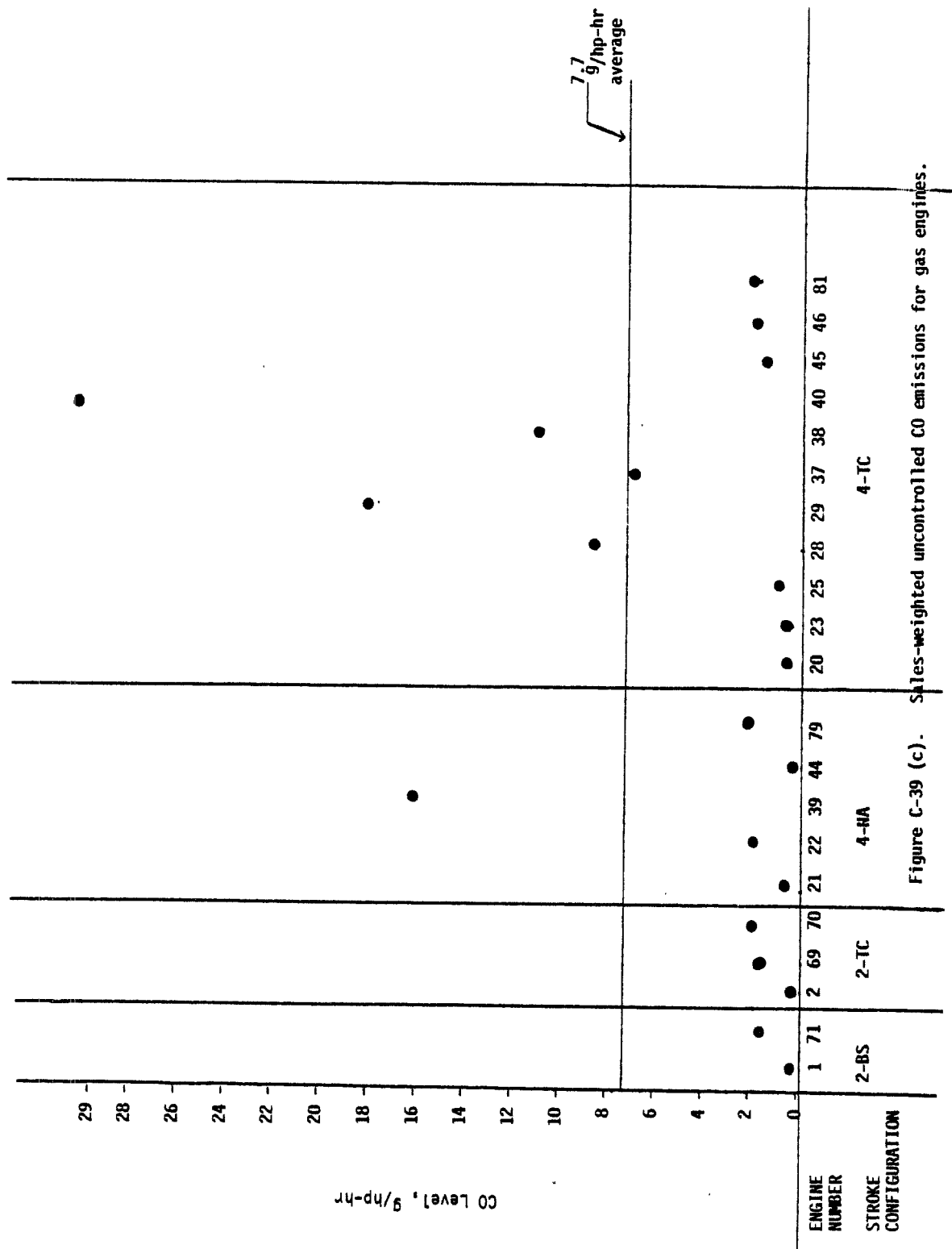


Figure C-39 (c). Sales-weighted uncontrolled CO emissions for gas engines.

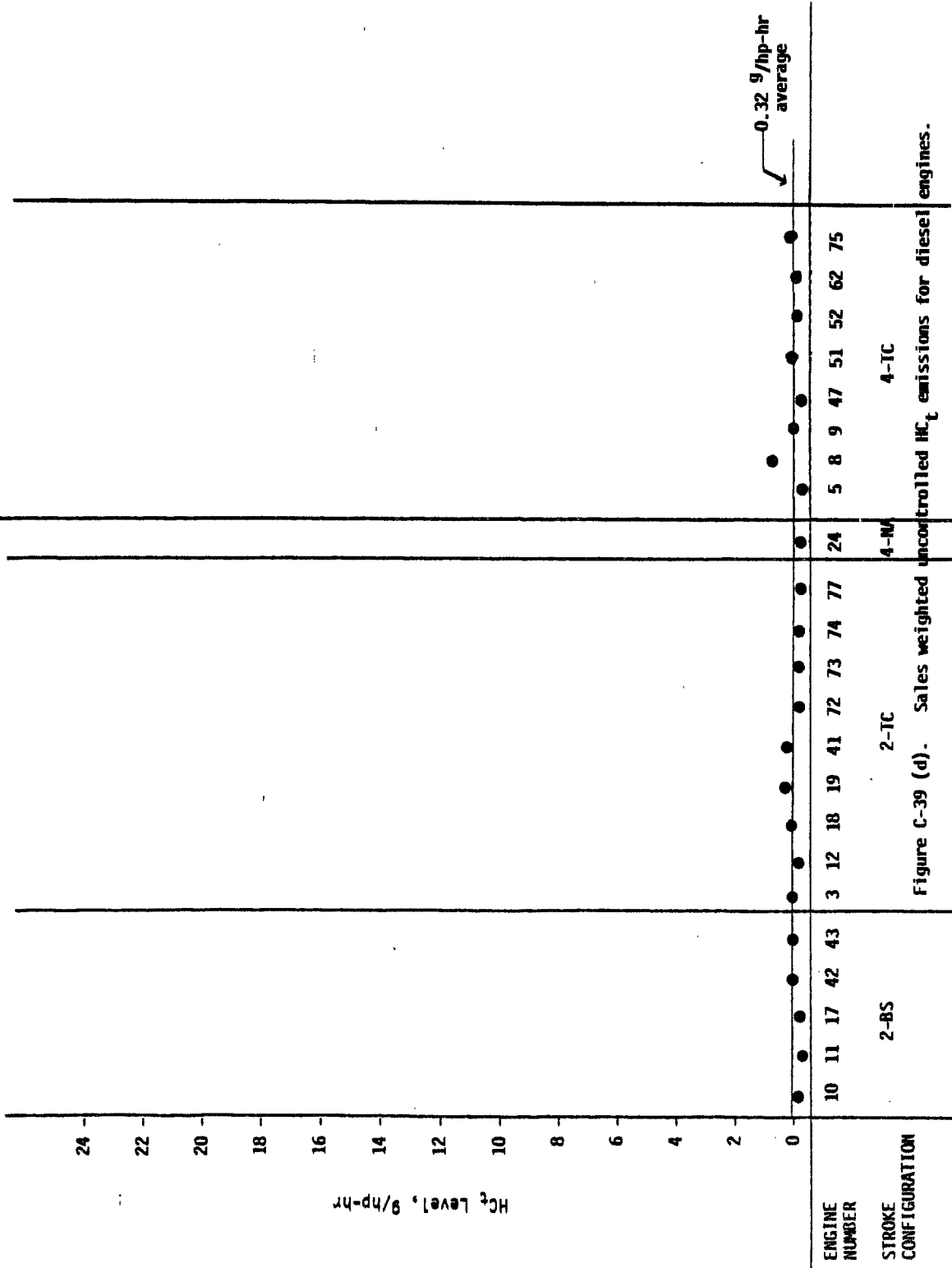
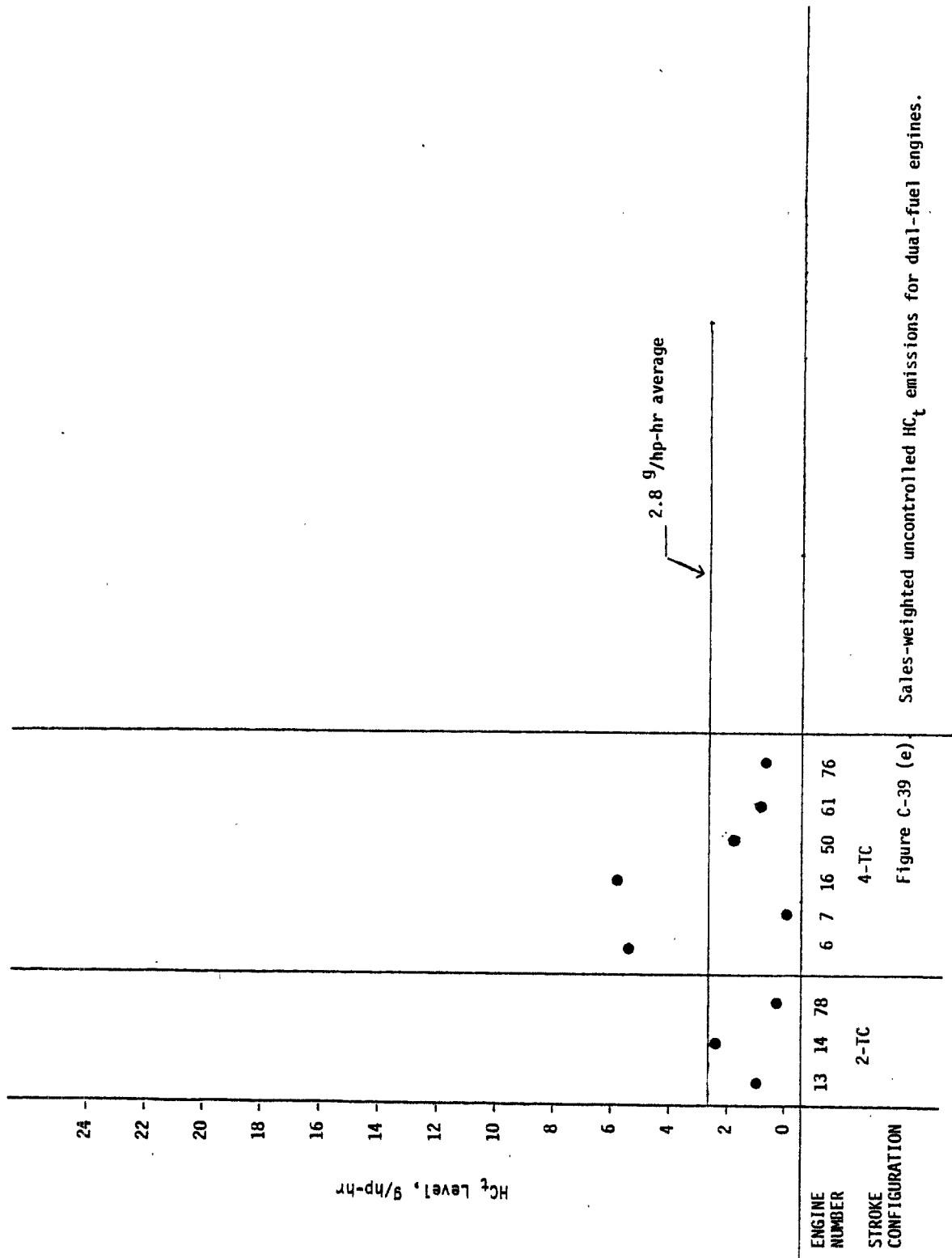


Figure C-39 (d). Sales weighted uncontrolled HC_t emissions for diesel engines.



Sales-weighted uncontrolled HC_t emissions for dual-fuel engines.

Figure C-39 (e)

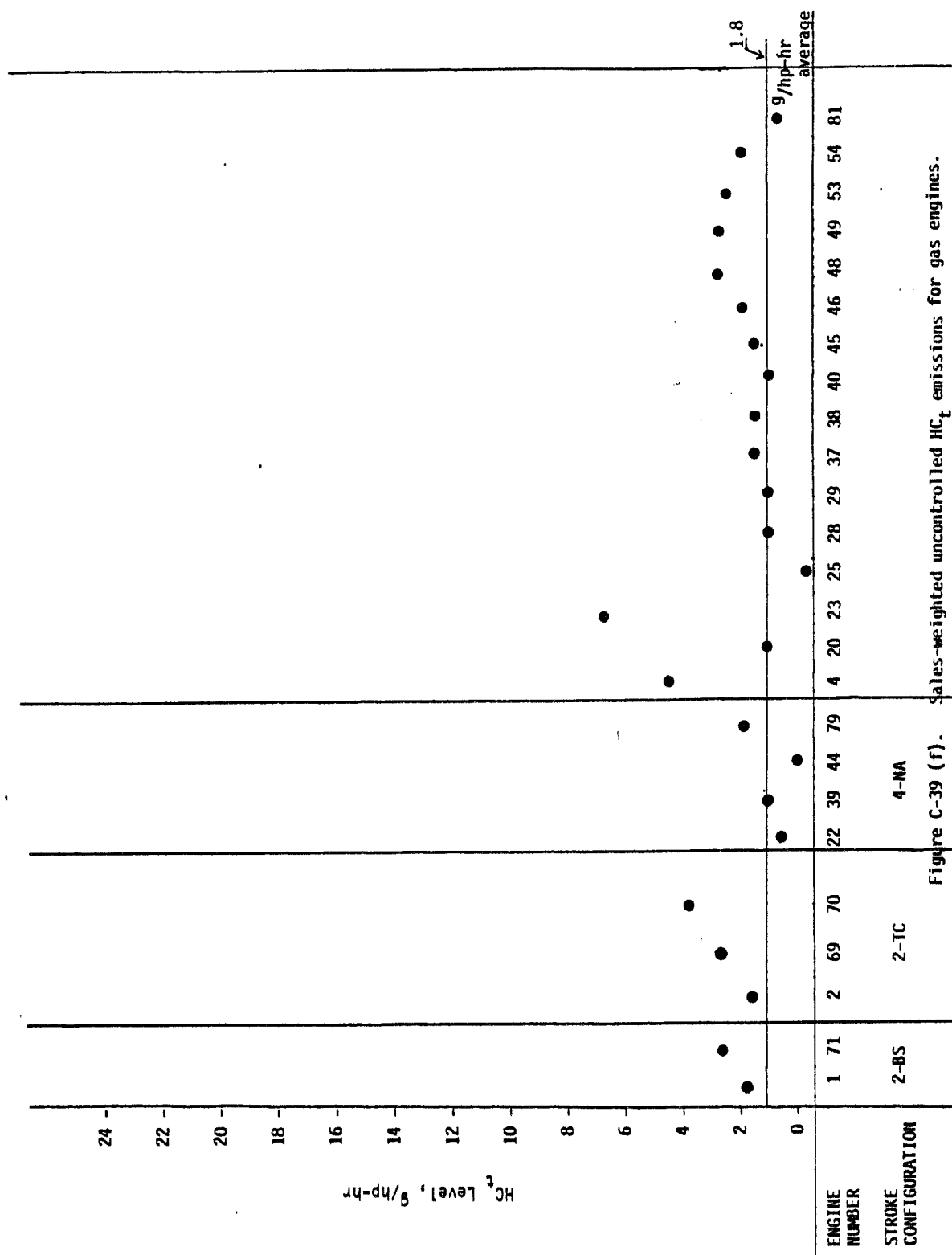


Figure C-39 (f). Sales-weighted uncontrolled HC_t emissions for gas engines.

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GLOSSARY

aftercooler	-- a heat exchanger used to cool inlet air that has passed through a turbocharger. Also referred to as an intercooler (see Figure 4-32).
air-cooled	-- a method of engine cooling, often used on small engines. Air is sucked or forced over the engine (by an engine-driven fan) transferring heat from the engine block. Fins are generally used to conduct heat away from the combustion chambers and increase the area available for heat transfer.
air injection	-- an emission control technique in which air is pumped (injected) into the exhaust manifold to complete the reaction of any unburned fuel and CO which is still present.
air-to-fuel ratio	-- a ratio of the mass flowrate (g/hr) of air into an engine to the mass flowrate of fuel (g/hr).
annualized cost	-- initial costs, or the sum of initial and operating costs, expressed as an annual charge.
baseload	-- continuous operation, generally considered to be 8000 hr/yr.
BDC	-- see bottom dead center.
blower scavenging	-- a method of charging the cylinder of an engine with air in which a low-pressure blower driven by the engine forces air into the cylinder.
bore	-- the diameter of the cylinder of an engine.
bottom dead center	-- position of the piston when it is at the bottom of the cylinder; corresponds to maximum available gas volume.
brake horsepower (bhp)	-- the power delivered by the engine shaft at the output end. The name is derived from the fact that it was first measured by the power consumed in a brake attached to the output shaft. Brake horsepower equals

	total power delivered by the pistons less losses and power used to drive auxiliaries (fan, pumps, etc.).
brake mean effective pressure (bmep)	-- the hypothetical constant pressure that would have to be exerted on the head of the piston during the entire stroke to generate the same torque as is actually generated by the engine. Hence, frequently used synonymously with torque.
brake specific	-- emissions expressed on the basis of power output, i.e., emissions g/hp-hr.
carburetor	-- a device on a spark ignition engine which controls the flowrate of air and fuel and mixes them in the proper proportions for combustion.
catalytic converter	-- a device which uses a catalyst to promote a reaction that alters the chemical composition of the gas passing through it. Oxidation type converters change CO and HC to CO ₂ and H ₂ O using a precious metal catalyst, such as platinum. Reduction catalysts are used to reduce NO to N ₂ and O ₂ .
cetane number	-- a reference number for compression ignition engine fuels. Higher numbers indicate better ignitability and better antiknock characteristics.
chemiluminescent analyzer	-- a device used to measure NO _x emissions.
compression ignition (CI)	-- one of two methods of initiating combustion in the engine cylinder. In CI engines, the air charge is introduced into the cylinder and compressed, thereby raising its temperature above the auto-ignition temperature of the fuel (temperature at which the fuel ignites spontaneously). Fuel is then injected into this hot compressed air and ignites spontaneously. All diesel and dual fuel engines are compression ignited.
compression ratio (CR)	-- ratio of the volume in the combustion chamber when the piston is at the bottom of the stroke to that when the piston is at the top.
connecting rod	-- a rod which connects the piston to the crankshaft and permits the reciprocating motion of the piston to be transferred to rotary motion by the crankshaft.
continuous rating	-- see rating.
cooling system	-- the system by which combustion heat is removed from engine block. This system may consist of a water jacket through which a liquid is circulated to remove

heat from the engine and a radiator, cooling tower, or heat recovery system to remove the heat from the liquid before it is recirculated through the water jacket. It may also consist of fins to conduct heat away from the combustion chambers and a fan to move air past these fins to remove the heat from them (see "air-cooled").

- cost effectiveness -- ratio of the cost to reduce a pollutant from an engine to the amount of pollutant removed (\$/kg).
- crankcase blowby -- unburned fuel, combustion gases, and lubricating oil which escape from the cylinder past the piston rings into the crankcase and are then vented.
- crankshaft -- the shaft that receives engine power from the reciprocating motion of the pistons and delivers it as rotary motion.
- crude -- unrefined oil.
- cylinder liner -- a steel liner inserted into the cylinder. It can be removed and replaced after excessive wear.
- denitrification -- the removal of nitrogen from fuel during refining. Some removal occurs concurrently with sulfur removal.
- derating -- a control technique which limits the maximum load of an engine to less than the design value.
- desulfurization -- the removal of sulfur from fuel. This occurs during the refining of crude oil into light distillates and gasoline.
- digester gas -- fuel gas formed from sewage sludge. Primary constituents are CH_4 (methane), CO_2 , and H_2 .
- direct injection -- the injection of fuel directly into the cylinder.
- displacement -- the volume a piston sweeps out moving from the bottom of the cylinder (bottom dead center, BDC) to the top of the cylinder (top dead center, TDC).
- distillate -- a product of the refining of crude oil. The most common distillates are No. 2 diesel oil and gasoline.
- dribble -- the loss of fuel from the injector tip after fuel injection.
- dry -- refers to gas measurements made under conditions where water has been removed from the gas before the measure is made. Also gas measurements which have been

mathematically adjusted to be equivalent to measurements made on water-free gas.

- durability -- the ability of an engine to operate throughout a normal service life without excessive wear or failure of engine components.
- exhaust gas recirculation (EGR) -- an emission control remote technique whereby a portion of the exhaust gases are retained in the cylinder (internal EGR) or are routed back to the engine intake (external EGR) to displace some of the inlet air.
- exhaust manifold -- an internally ducted casting which receives exhaust products from the cylinders and transfers these products to the exhaust systems (see Figure 4-47).
- field gathering -- collection of oil or natural gas at the surface of a well (the well-head) into the pipes (feeder lines) which carry it to the major pipelines.
- flame ionization detector (FID) -- an analytical instrument used to measure HC emissions.
- fossil steam -- steam for electric utility power generation produced from the combustion of coal, oil, or natural gas.
- four-stroke -- a type of engine which requires four traverses of the piston in the cylinder (two revolutions of the crankshaft) per power stroke (i.e., to complete a cycle).
- fuel additive -- a substance added to fuel, usually to reduce smoke from an engine.
- fuel-bound nitrogen -- nitrogen contained in the fuel rather than in the air.
- fuel pump -- device which pumps fuel to the fuel injection system or carburetor.
- governor -- device which controls the amount of fuel supplied to the cylinder according to the load demand on the engine, e.g., a governor is used to maintain a given speed (rpm) under varying load in electric generator applications.
- heavy duty mobile -- a term referring to vehicles over 6000-lbs gross vehicle weight (total weight when vehicle is fully loaded) subject to federal emission standards.
- higher heating value -- the heat produced by the complete combustion of a unit quantity of fuel (at standard conditions of temperature, pressure, and humidity) such that the

water in the products of combustion is in the liquid phase.

- horsepower -- the time rate of doing work. One U.S. horsepower is equal to 33,000 foot-pounds per minute. (One U.S. horsepower equals 1.014 metric horsepower.)
- indirect injection -- the injection of fuel into a precombustion chamber, or antechamber, where combustion commences in an oxygen deficient environment before expanding into the cylinder for completion of the combustion in the presence of excess oxygen (see Figure 4-42).
- initial cost -- the purchase price of the engine, including all auxiliaries necessary for its operation (e.g., radiator, lube oil pump, aftercooler, starting motor, mounting skid, etc.).
- injection rate -- the rate at which fuel is introduced into the cylinder by the injector.
- injection timing -- the time, measured in degrees of crankshaft rotation, that it takes for fuel to be admitted into the cylinder. Start of injection (in degrees before TDC) is an important parameter in emissions control.
- injector -- device which injects fuel into the cylinder (see Figure 4-55).
- injector rack -- a mechanical linkage controlled by the governor which determines the pressure of the fuel supplied by the fuel pump to the fuel injector.
- intake manifold -- internally ducted casting that distributes incoming air or air and fuel mixture into the cylinders (see Figure 4-32).
- intercooler -- see aftercooler.
- intermittant rating -- see rating.
- jacket water -- engine cooling water that circulates from a cooling tower or radiator through cavities (or jackets) in the engine block.
- knock -- premature ignition of fuel in cylinder of an engine.
- lean -- refers to an air and fuel mixture which contains more air than stoichiometrically necessary to completely burn all the fuel.

low sac nozzle injector -- a fuel injector with a minimum of volume beneath the injector nozzle, thus reducing injector dribble and HC emissions (see Figure 4-55).

lower heating value -- the heat produced by the complete combustion of a unit quantity of fuel (at standard conditions of temperature, pressure, and humidity) such that the water in the products of combustion is in the vapor phase.

major overhaul -- an overhaul which generally includes removal and replacement (or reconditioning) of the cylinder liners, pistons, rings, valves, fuel pump, and crankshaft. The engine is frequently removed from the installation to perform this overhaul.

manifold air temperature cooling (MAT) -- a control technique based on lowering the intake air temperature. Usually accomplished with an intercooler (see Figure 4-32).

maximum rating -- see rating.

minor overhaul -- an overhaul which generally includes replacing rings, valves, injectors or spark plugs, and occasionally pistons. This overhaul is usually performed without removing the engine from its installation and at more frequent intervals than major overhauls.

naturally aspirated -- a type of air charging whereby the piston draws air into the cylinder as it travels to the bottom of the cylinder. Not turbocharged, supercharged, or blower-scavenged.

nondispersive infrared analyzer (NDIR) -- an instrument used to measure CO and CO₂.

nonpropulsive -- an application where the engine is never used to move the structure or device on which it is mounted.

nuclear steam -- steam (usually for electric generation) produced by heat from nuclear reactions.

operational change -- a change in engine operation which requires only an adjustment of given operating conditions but no hardware additions -- e.g., ignition timing.

original equipment manufacturer (OEM) -- a firm which, in this case, buys an engine from an engine manufacturer and incorporates it into a product of which the engine is only a component -- e.g., a portable compressor, which is usually assembled by an

OEM using his own housing and compressor but a purchased engine.

- overhaul -- the replacement of worn parts in an engine, usually classified as either a minor or major overhaul depending upon how many, and which, parts are replaced. (See minor and major overhaul.)
- oxidation -- when used in this document, oxidation refers to the addition of oxygen to a molecule by a chemical reaction, as when CO combines with oxygen to form CO₂. (The more general definition is the loss of electrons by a reactant in a chemical reaction.)
- paramagnetic analyzer -- an instrument used to measure oxygen content of exhaust.
- peak cylinder temperature -- the maximum temperature in the cylinder during combustion.
- pilot oil -- a small amount (<10 percent by heating value) of diesel fuel injected near the end of the compression stroke of a dual-fuel engine to ignite the gaseous fuel.
- piston crown -- the upper surface of the piston.
- piston ring -- a thin ring of metal which fits into a groove in the piston. Piston rings seal the combustion volume by filling the gap between the piston and the liner.
- ports -- openings in the cylinder liner through which air and fuel enter the cylinder and/or combustion gases leave it. Used in two-stroke engines.
- precombustion chamber -- see indirect injection.
- physical change -- a hardware change to an engine (as opposed to an operational change).
- quenching -- the cooling below the combustion point of fuel which has impinged on the walls of the cylinder.
- rating -- brake horsepower output of the engine. Reported either as continuous (power that engine can deliver continuously), intermittent (power that engine can deliver intermittently, usually for no more than 1 hour during a consecutive 2-hour period), or maximum (peak power than engine can deliver). Continuous ratings are used in this document.

reactivity (photochemical)	-- potential of a hydrocarbon compound to react with other species in the atmosphere to produce smog.
reduction	-- when used in this document, reduction refers to the removal of oxygen from a molecule by a chemical reaction, as when NO is reduced to N ₂ and O ₂ in the presence of a catalyst. (The more general definition is the gain of electrons by a reactant in a chemical reaction.)
residence time	-- the time interval after ignition during which the air-fuel mixture can burn at elevated temperature and pressure.
residual	-- a heavy viscous oil, often containing large amounts of sulfur (>1 percent by volume), which remains after distillation to produce other fuels.
retard	-- a NO _x control technique wherein ignition of the fuel is delayed by delaying the spark (SI engines) or the start of fuel injection (CI engines) by several degrees of crankshaft rotation.
rich	-- refers to an air and fuel mixture which contains less oxygen than is stoichiometrically necessary to completely burn all the fuel.
rpm	-- rotations per minute; a measure of the engine crankshaft rotational speed.
sac	-- the small volume below the nozzle of an injector (see Figure 4-55).
scrubber	-- a device which removes a pollutant from an exhaust gas stream through absorption of the pollutant by the scrubber liquid.
smoke	-- visible emissions from an engine exhaust.
spark ignition (SI)	-- one of two methods of initiating combustion in the engine cylinders (see also compression ignition). In SI engines, an electrical spark is generated across a gap between two electrodes at the tip of the spark plug to ignite the fuel-air mixture. All gasoline and natural gas fueled engines are spark ignited.
spark timing	-- the degrees of crankshaft rotation before top dead center (TDC) at which the spark commences.
squish lip	-- refers to a cavity shape in the piston head which generates squish, or radially outward motion of the air-fuel mixture. Combustion is initiated in this cavity.

standby	-- a limited usage service, typically 200 hr/yr or less (e.g., emergency electrical generators).
stationary	-- an application in which the engine is never used to propel the structure or device on which it is mounted.
stoichiometric	-- refers to the chemically correct amount of air (oxygen) to completely burn a given amount of fuel.
stratified charge	-- the staged combustion of fuel, first partially under rich conditions, and completely later at cooled temperatures in the presence of excess air (see Figures 4-44(a) and 4-44(b)).
stroke	-- the vertical distance that the piston travels between the bottom and the top of the cylinder. Also the movement of the piston between these points.
swirl	-- rotational motion of the air or air-fuel mixture in the cylinder. Used mainly to enhance air-fuel mixing. Can be low, medium, or high swirl depending on ratio of rotational speed to average piston speed.
TDC	-- see top dead center.
thermal efficiency	-- ratio of the engine energy output to the energy input based on the lower heating value of the fuel used.
thermal fixation	-- the formation of NO_x by reaction of nitrogen in the inlet air with oxygen during the combustion process.
thermal reactor	-- a device which combines air with the unburned fuel and CO in the exhaust to promote further combustion of these gases, thereby reducing HC and CO emission.
top dead center	-- position of the piston when it is at top of the cylinder; corresponds to minimum available gas volume.
torque	-- a movement (the product of a force and a lever arm) which produces rotation.
turbocharger	-- a device which uses a centrifugal compressor to increase the pressure of the incoming air. The compressor is powered by an exhaust gas driven turbine. It is used to increase the power output and efficiency of the engine (see Figure 4-32).
two-stroke	-- a type of engine which requires two traverses of the piston in the cylinder (one revolution of the crankshaft) per power stroke (i.e., to complete a cycle).

- valve -- a device which opens into the cylinder to admit air or air and fuel mixture or to exhaust combustion products.
- valve camshaft -- a shaft driven off the crankshaft having eccentric lobes which open the intake and exhaust valves at the proper time.
- valve overlap -- the interval, in degrees of crankshaft rotation, during which the intake and exhaust valve are both open.
- volatility -- a measure of the ability of a fuel to evaporate at a given temperature.
- wet -- refers to the existence of water vapor in an exhaust gas sample from water of combustion and water contained in the intake air.

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16. ABSTRACT Standards of performance to control nitrogen oxides from new, modified and reconstructed stationary internal combustion engines in the U. S. are being proposed under section 111 of the Clean Air Act. This document contains information on the internal combustion engine industry and emission control technology, a discussion of the selected emission limits and the supporting data and the alternatives which were considered, and analyses of the environmental and economic impacts of the proposed standards.		
17. KEY WORDS AND DOCUMENT ANALYSIS		
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